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The Conundrum of the Power Plant

By Berton Braley

(With apologies to Kipling)

When the smooth-tongued salesman comes around with his talk of some new device,
Which he says is the niftiest thing on earth and worthy of twice its price,
He may be telling the gospel truth as clear as the light of day,
But the wise man mutters behind his hand, "It looks good, but will it pay?"

For the test of a grate, a metering device or recorder of CO_2
Is never its looks or the salesman's talk, but rather what it will do;
So the wise man gazes at plots and prints and hears what the salesmen say,
But quietly figures the costs and sighs, "It looks good, but will it pay?"

Or the man from the central station comes, and a plausible chap is he,
And he wraps himself in a mist of words like a roseate rhapsody;
He would junk your plant, and he'd sign you up on a contract right away,
But the wise man murmurs in accents low, "It looks good, but will it pay?"

Efficiency experts dope things out on the latest and smartest plan,
They count the strokes of each piston rod and the motions of every man;
And they're frequently right in their scheme of things and the logic that they display,
But the wise man nevertheless exclaims, "It looks good, but will it pay?"

Now there's room for progress in every plant, and the mossback's just a fool,
And the man who balks at the thought of change is worse than a stubborn mule;
But the "show me" chap has the right idea and never goes far astray
When he puts each change to the acid test—"It looks good, but will it pay?"

Hand-Firing Soft Coal

By Charles H. Bromley

SYNOPSIS—In the main the article is a presentation of the best methods of building and maintaining a soft-coal fire in a hand-fired furnace. A hand-fired setting excellently suited for smokeless and economical combustion is shown, and the behavior of the volatile gases is described.

If you look at a map of the coal fields of the United States, you will with difficulty find the anthracite areas, so small are they in comparison with the bituminous and lignite fields. This means that the consumption of soft coal must rapidly increase in localities where anthracite is now the major fuel. It is the duty of engineers and firemen to smokelessly and economically burn the fuel—first because combustion is part of their everyday work, and second, because personal and civic pride demands that they do their best to beautify or to preserve the beauty of the community. In this article bituminous coal is considered exclusively, and the best methods of burning it in hand-fired horizontal tubular boilers as demonstrated under commercial conditions. However competent the man, smokeless and economical combustion is commercially impossible unless the furnace is suitable; that is, has volume, proper proportions and adjuncts. Without these essentials—namely, draft, temperature, time, and gas and air mixing provisions—good combustion cannot be had.

A typical soft coal has the following principal constituents as received: Fixed carbon, 57.6 per cent.; volatile, 32.24 per cent.; ash, 7.32 per cent.; moisture, 2.84 per cent. The coal is a solid. On being burned everything but the ash and what combustible remains in it is driven off as gas accompanied by a small quantity of fine solid particles. The volatile matter consists largely of oils which will distill off, some at temperatures a little below 300 deg. F., and others, the heavier ones, at 800 to 950 deg. F. Running normally, the furnace temperature will be about 1,800 to 2,500 deg. F. (hand-firing)—from six to eight times as high as needed to support dis-

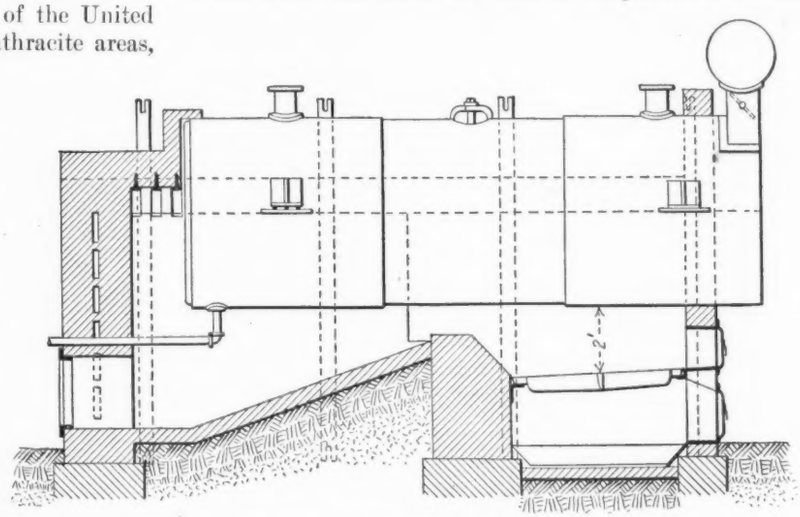


FIG. 1. THE OLD HARTFORD STANDARD SETTING

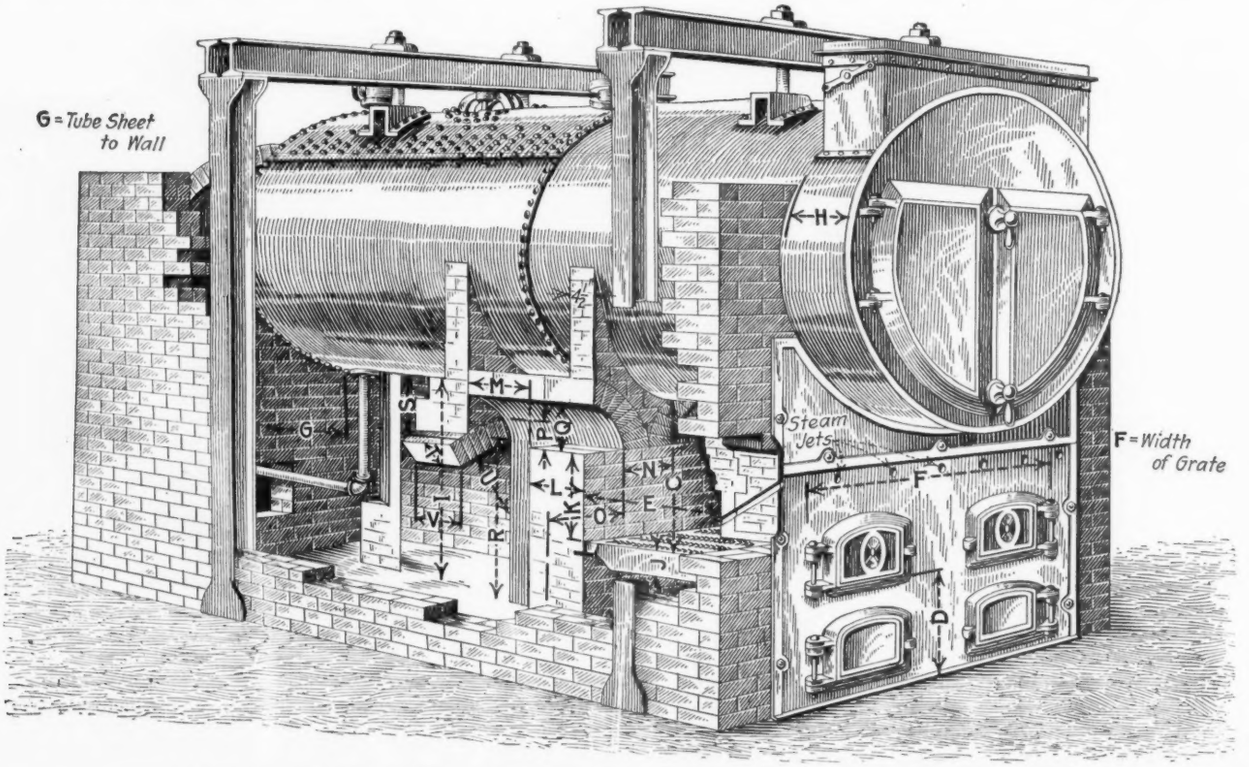


FIG. 2. EXCELLENT TYPE OF SETTING APPROVED BY MOST SMOKE INSPECTORS

tillation. After these oils vaporize and gasify, they must receive enough air for their combustion, must have sufficient temperature to carry on combustion and must be well mixed with the air for combustion. And all this

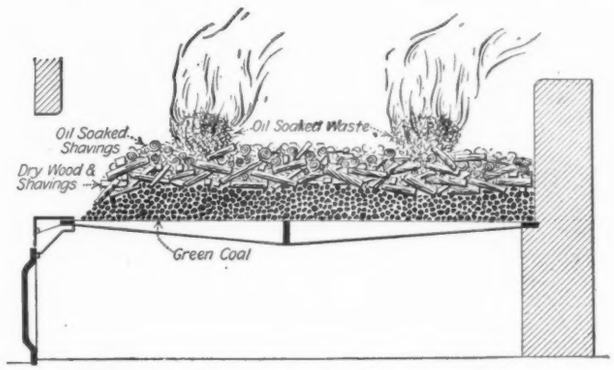


FIG. 3. STARTING THE FIRE

should happen before the gases get into the boiler tubes; in fact, for best results the whole combination of events should be completed before the gases touch the heating surface at all—in less than a second, even with an uncommonly high setting. Is it to be wondered at, then, that this is impossible in the widely used Hartford standard setting, Fig. 1, where the boiler shell is but two feet above the grate?

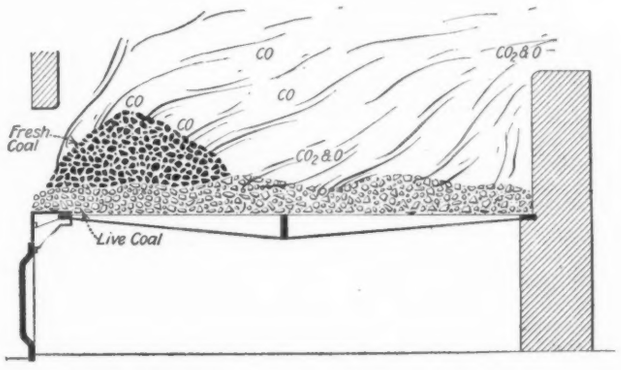


FIG. 4. COAL COKING AT THE DEADPLATE

When Osborne Monnett tackled the job of cleaning Chicago of smoke, he found that over 90 per cent. of the boilers were of the horizontal return-tubular type having settings wherein the distance from grate to shell was not more than two feet. This condition is true of most cities, but fortunately is becoming less and less, owing to the vigor of smoke abatement campaigns.

To briefly state a long story, the flush-front and the extension-front dutch oven and several other types of settings were tried, and that which gave the best results, smokelessness and economy considered, is shown in Fig. 2. This setting is recommended by nearly all city smoke inspectors and is becoming standard in soft-coal burning districts. It represents the application of facts discovered and corroborated by thousands of tests and observations. The dimensions as given in the table accompanying Fig. 2 are those used by the Smoke Inspection Department of Cincinnati and practically the same as those recommended by Chicago. The writer recommends that for all hand-fired furnaces using coal above 25 per cent. volatile the distance from shell to grate be not less than 4 ft. or the nearest approach to that height that commercial brick will give.

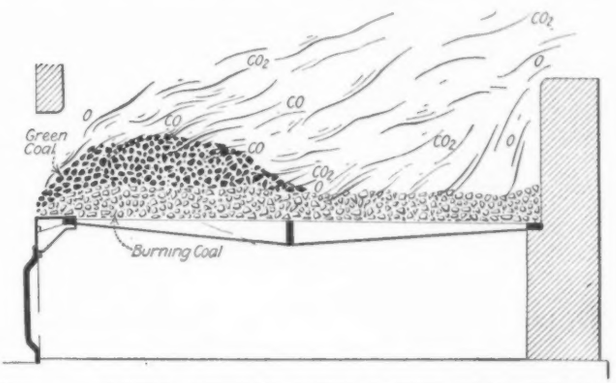


FIG. 5. COKING METHOD WITH FRESH COAL SPREAD OUT MORE THAN IN FIG. 4

SPECIFICATIONS FOR IMPROVED SETTING HORIZONTAL TUBULAR BOILERS TO ACCOMPANY FIG. 2

Diameter shell	84 in.	84 in.	78 in.	78 in.	72 in.	72 in.	66 in.	66 in.	60 in.	60 in.	54 in.	54 in
Length shell	20 ft.	18 ft.	20 ft.	18 ft.	18 ft.	16 ft.	18 ft.	16 ft.	18 ft.	16 ft.	16 ft.	14 ft.
C	42 in.	42 in.	42 in.	42 in.	36 in.	36 in.	36 in.	36 in.	36 in.	36 in.	36 in.	36 in.
D	2 ft. 2 in.	2 ft. 2 in.	2 ft. 2 in.	2 ft. 2 in.	2 ft.	2 ft.	2 ft.	2 ft.	1 ft. 10 in.	1 ft. 10 in.	1 ft. 10 in.	1 ft. 10 in.
E	6 ft. 6 in.	6 ft.	6 ft. 6 in.	6 ft.	6 ft.	5 ft. 6 in.	5 ft. 6 in.	5 ft.	5 ft. 6 in.	5 ft.	4 ft. 6 in.	4 ft.
F	7 ft.	7 ft.	6 ft. 6 in.	6 ft. 6 in.	6 ft.	6 ft.	5 ft. 6 in.	5 ft. 6 in.	5 ft.	5 ft.	4 ft. 6 in.	4 ft. 6 in.
G	2 ft. 10 in.	2 ft. 10 in.	2 ft. 8 in.	2 ft. 8 in.	2 ft. 6 in.	2 ft. 6 in.	2 ft. 4 in.	2 ft. 4 in.	2 ft. 2 in.	2 ft. 2 in.	2 ft.	2 ft.
H	1 ft. 10 in.	1 ft. 10 in.	1 ft. 9 in.	1 ft. 9 in.	1 ft. 8 in.	1 ft. 8 in.	1 ft. 7 in.	1 ft. 7 in.	1 ft. 6 in.	1 ft. 6 in.	1 ft. 5 in.	1 ft. 5 in.
I	6 ft. 4 in.	5 ft. 10½ in.	6 ft. 5½ in.	6 ft.	5 ft. 11½ in.	5 ft. 6½ in.	5 ft. 7 in.	5 ft. 1½ in.	5 ft. 9 in.	5 ft. 4 in.	4 ft. 11 in.	4 ft. 8 in.
J	6½ in.	6 in.	6½ in.	6 in.	6 in.	5½ in.	5½ in.	5 in.	5½ in.	5 in.	4½ in.	4 in.
K	1 ft. 10½ in.	2 ft.	1 ft. 9½ in.	1 ft. 11 in.	1 ft. 6 in.	1 ft. 7 in.	1 ft. 7 in.	1 ft. 8 in.	1 ft. 5½ in.	1 ft. 6½ in.	1 ft. 7½ in.	1 ft. 6 in.
L	2 ft. 3 in.	2 ft. 3 in.	2 ft. 3 in.	2 ft. 3 in.	2 ft.	2 ft.	1 ft. 10½ in.	1 ft. 10½ in.	1 ft. 10½ in.	1 ft. 10½ in.	1 ft. 6 in.	1 ft. 6 in.
M	3 ft.	3 ft.	2 ft. 10 in.	2 ft. 10 in.	2 ft. 6 in.	2 ft. 6 in.	2 ft. 4 in.	2 ft. 4 in.	2 ft. 2 in.	2 ft. 2 in.	2 ft.	2 ft.
N	1 ft. 6 in.	1 ft. 6 in.	1 ft. 6 in.	1 ft. 6 in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.
O	2 ft. 9 in.	2 ft. 9 in.	2 ft. 6 in.	2 ft. 6 in.	2 ft. 5½ in.	2 ft. 5½ in.	2 ft. 2½ in.	2 ft. 2½ in.	1 ft. 11½ in.	1 ft. 11½ in.	1 ft. 8½ in.	1 ft. 8½ in.
P	1 ft. 4 in.	1 ft. 2 in.	1 ft. 5½ in.	1 ft. 3½ in.	1 ft. 1 in.	1 ft. 1½ in.	1 ft. 1½ in.	1 in.	1 ft. 3 in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 2½ in.
Q	2 ft. 5½ in.	2 ft. 3½ in.	2 ft. 5½ in.	2 ft. 3½ in.	2 ft. 2 in.	2 ft. ½ in.	2 ft. ½ in.	1 ft. 11 in.	2 ft. ½ in.	1 ft. 11 in.	1 ft. 8½ in.	1 ft. 9 in.
R	4 ft. 2 in.	3 ft. 10½ in.	4 ft. 2½ in.	3 ft. 11 in.	3 ft. 11½ in.	3 ft. 8 in.	3 ft. 8½ in.	3 ft. 4½ in.	3 ft. 9 in.	3 ft. 5½ in.	3 ft. 2 in.	2 ft. 10 in.
S	2 ft. 2 in.	2 ft.	2 ft. 3 in.	2 ft. 1 in.	2 ft.	1 ft. 10½ in.	1 ft. 10½ in.	1 ft. 9 in.	2 ft.	1 ft. 10½ in.	1 ft. 9 in.	1 ft. 10 in.
T	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.	9 in.
U	1 ft.	1 ft.	1 ft.	1 ft.	1 ft.	1 ft.	11 in.	11 in.	10 in.	10 in.	9 in.	9 in.
V	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1¼ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.	1 ft. 1½ in.
Number and size of tubes	92–4 in.	92–4 in.	88–4 in.	88–4 in.	70–4 in.	70–4 in.	54–4 in.	54–4 in.	46–4 in.	46–4 in.	36–4 in.	36–4 in.
Heating surface in tubes	1932.0	1738.8	1842.8	1658.5	1323.0	1176.0	1017.7	904.6	866.9	770.6	604.8	529.2
Heating surface in shell	219.9	197.9	204.2	183.8	169.6	150.7	155.5	138.2	141.3	125.6	113.1	99.0
Total heating surface tubes and shell, sq.ft	2151.9	1936.7	2047.0	1842.8	1492.6	1326.7	1173.2	1042.9	1008.2	896.2	717.9	628.2
Boiler horsepower	215	194	205	184	149	133	117	104	101	90	72	63
Square feet of grate surface	45.5	42.0	42.25	39.0	36.0	33.0	30.25	27.5	27.5	25.0	20.25	18.0
Area through each bridge wall retort, sq.in	820.0	756.0	761.0	702.0	648.0	594.0	544.5	495.0	495.0	450.0	364.5	324.0
Area between bridge wall and deflection arch, sq.in.	2448.0	2448.0	2040.0	2040.0	1770.0	1770.0	1484.0	1484.0	1326.0	1326.0	984.0	984.0
Area under deflection arch	3280.0	3024.0	3041.0	2808.0	2592.0	2376.0	2177.0	1979.0	1979.0	1800.0	1457.0	1296.0
Area of tubes, sq.in	1011.0	1011.0	967.0	967.0	769.0	769.0	593.0	593.0	505.0	505.0	396.0	396.0
25% over flue area	1264.0	1264.0	1208.0	1208.0	961.0	961.0	741.0	741.0	631.0	631.0	495.0	495.0

Notes—First grade of firebrick to be used throughout with the exception of the combustion chamber floor and the side walls of the chamber back from a point one foot behind the rear face of the deflection arch. Fire-doors MUST provide for special air admission of an area equal to 4 sq.in. for each square foot of grate surface, when working pressure on boiler is 20 pounds or less. If desired, these doors may be used on boilers carrying more than 20 pounds pressure. Make the effective area through damper frame not less than the combined area of tubes. See ordinance for breeching and other regulations. Table applies only for the ordinary conditions and is subject to revision or modification for unusual conditions. Heating surfaces are in square feet, areas in square inches, except for grate surface.

To build a fire in a cold boiler without making objectionable, or, to be specific, say No. 3 Ringlemann smoke, is regarded as at least extremely doubtful by the average man. But it can be done. Here is the way H. H. Maxfield, superintendent of motive power, Pennsylvania

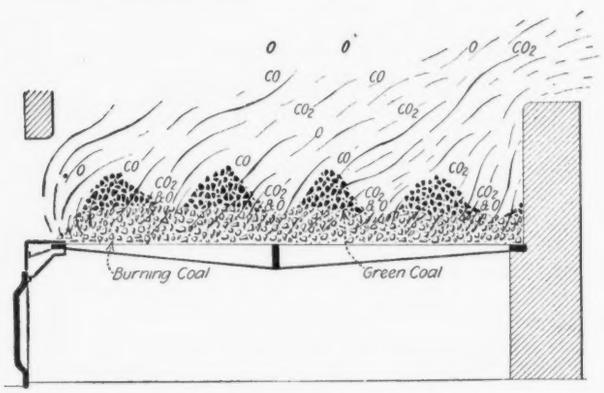


FIG. 6. COAL COKING IN SMALL PILES

R.R., Western Pennsylvania Division, Pittsburgh, does it in locomotives, and the method is applicable to stationary boilers, perhaps with better results for the same skill applied.

The entire grate is covered with 3 in. of gas coal (assumably a coal containing from 32 to 38 per cent. volatile) spread evenly, and upon this is laid a layer of low-volatile coal (about 20 per cent.) thoroughly wet to a depth of 4 in. Dry wood shavings are then spread over the surface of the coal, after which an ordinary bucket of shavings soaked in crude oil is distributed on top of the dry shavings. A piece of oil-soaked waste is thrown into the middle of the grate to ignite the shav-

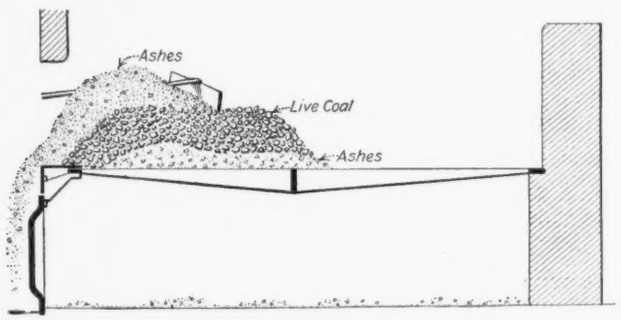


FIG. 7. BACK HALF OF GRATE CLEANED FIRST

ings (see Fig. 3). The blower is opened very lightly and the opening increased as the coal becomes ignited. The fire-door is kept open until after the coal is burning, or longer, depending upon the condition of the stack. Sometimes more coal is needed before a good fire is had, and if so, wet low-volatile coal is used. The gas coal is used to prevent the low-volatile coal from falling through the grate.

This method of building a fire is correct from every viewpoint. The fire burns from the top down, causing the volatile to pass through a zone hot enough to completely gasify the vapors. The fire should not be sliced or hooked or otherwise disturbed. The method is applicable to a stationary boiler, and although the low-volatile coal may not be easily had, coal from the everyday supply will do. Fig. 3 shows the method applied to a stationary boiler.

Maintaining the fire between cleaning periods requires no mean skill. The coking method with which every fireman who reads should be familiar is not always the best. A glance at Fig. 4 shows that only a small part of the total surface of the green coals coking is exposed to the direct heat of the furnace. If the pile of coal coking could be left there long enough, the volatile would be driven off so slowly as to make no objectionable smoke. But it must soon be pushed and spread over the hot fire, and when this is done the volatile in the coal beneath the surface of the pile has reached a temperature where only a slight increase is needed to make it give forth its vapors so suddenly as to not give time for adequate mixing with air, thus making dense black smoke. The lowered furnace temperature also helps to make smoke at this time. All this is clear when we realize that the rapidity of distillation depends, for any given temperature, upon the surface of coal exposed to the heat. Less smoke will be made, even with a "puffy"

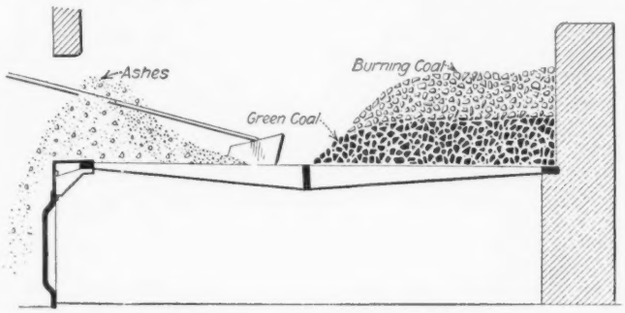


FIG. 8. CLEANING THE FRONT HALF OF THE GRATE

coal if the pile is spread as in Fig. 5, and if steam-air jets or other means of getting air in over the fire are used.

The alternate method of covering has the same disadvantage, in that the green coal, even though thrown on but one-half or one-third of the fuel bed, has so much of its surface exposed at once to the incandescent coal that distillation is more rapid than can be cared for by the air admission. Unfortunately, too, these vapors should receive heat as they go on their way to the uptake instead of losing it. With this method the gases do not

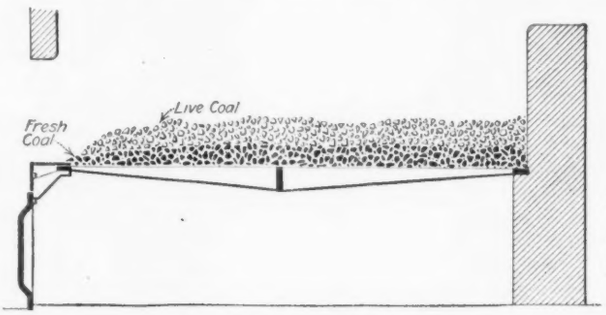


FIG. 9. THE FIRE CLEANED

pass directly over an incandescent area, as they do when the coking method is employed. Someone, I forget who, has suggested a way to cover in which the green coal is thrown in small piles well distributed over the burning fuel, and later, when most of the volatile has been distilled, these piles are leveled (see Fig. 6). The idea is to gradually expose the whole charge of coal to the direct

heat of the furnace. Theoretically the method is good, but it is hard work for the fireman and unfortunately gets him into the habit of disturbing the fuel bed with rake or slice bar, a most egregious fault.

Whether the coking, alternate, or the divided method of covering is used, or if the whole fire surface is covered at once, the covering must be thin and done frequently. This is particularly true of caking coals. Caking coals form the surface crust so peculiar to them during the coking process. The crust prevents sufficient air reaching the coal, and high spots form in the fuel bed. If these places are covered with fresh coal before the surface of the high spots have been broken and allowed to burn away, the trouble grows worse. Do not cover unburned places in the fuel bed. The troubles had with many caking coals may be made nearly negligible if the fire is always covered lightly and often. Use the rake and slice bar as little as possible, but remember that the crust of caking coal must be often broken, if formed, to maintain the desired steam pressure. When breaking the crust, be careful not to so disturb the fuel bed as to get ashes on or near the top of the fire. Usually this causes serious clinker trouble, which stops the normal flow of air through the fuel bed and gives rise to more caking.

For a not too-heavy covering and at normal hand-firing combustion rates, it takes from 2 to 6 min. for the volatile to be driven off. During the major part of this time use the steam jet if provided or open the fire-door to admit air in above the fire. The time it should be left open depends, of course, upon the rapidity of distillation and the quantity of the vapor and gas distilled.

From the smoke standpoint the difficult time to prevent smoke is during the cleaning period. First be sure that there is sufficient live coal when cleaning begins.

There is probably no more practicable satisfactory way of cleaning a fire than that shown in Figs. 7, 8 and 9. The back half of the grate is first cleaned as shown in Fig. 7, then about 3 in. of green coal is thrown on the bare grate and all the live coal pushed back on top of the green coal, which begins to burn from the top down, making it necessary for the vapors and gases to pass through the hot coal (see Fig. 8). The ashes are pulled from the front of the grate, green coal thrown on and the live coals at the back pulled forward and evenly distributed over the whole fuel-bed surface, Fig. 9.

Commutation in Continuous-Current Machines

By A. M. Bennett

SYNOPSIS—An elementary description of the process of commutating the current in the armature of a direct-current machine and the methods employed to obtain sparkless commutation.

An armature revolving in a field of alternate north and south poles has generated in its conductors an alternating electromotive force, owing to the fact that these conductors in passing adjacent pole faces are under the influence of lines of force having opposite polarity. The current flowing in a conductor will therefore change in direction every time it passes a pole; that is, there will be an alternation for each pole passed. During one alternation the electromotive force will rise in value from zero to maximum while the conductor passes from a point midway between two poles to a point opposite the middle of a pole, and will sink again to zero as it continues in its revolution and passes from under the pole to a position again between the poles.

Fig. 1 shows a coil arranged to revolve between the poles N and S; the ends of the coil are connected to slip rings on which brushes rest for connecting to the external circuit. Here are the essentials for the generation of current, which it must be remembered is always alternating in character within the armature windings, irrespective of whether the generator is of the so-called alternating-current or direct-current type. The coil consists of a single turn, but it is evident that any number of turns consistent with the requirement of design may be used in the same coil by extending one end around and alongside the other turn before connecting it to the slip ring.

Commutation is the action of transforming the alternating current generated as described, into continuous current, and it is the function of the commutator to effect this transformation. If in Fig. 1 a single ring cut in half is substituted for the two slip rings with one end of the coil connected to each half-ring, as shown in Fig. 2, an example of the commutator in its simplest form is obtained. Brushes bearing on the commutator serve as a connection to the external circuit. Such a device will transform the alternating current of the armature to current in one direction. Because of the single coil and the varying rate at which it cuts the flux, the current will be pulsating in character; that is, it will rise from zero to maximum and sink again to zero, for each pole passed by the armature coil. The induced electromotive force will be represented by the curve shown in Fig. 3.

In practice, to render the electromotive force of constant value instead of pulsating as shown, the armature coils and commutator segments are greatly increased in number. Fig. 4 is a diagram representing an armature with eight coils and commutator segments, the armature coils being shown wound on a ring after the manner of a Gramme armature, for the sake of clearness. The ending of one coil and the beginning of the next are connected to the same commutator segment, and this relation is preserved in connecting all coils to the commutator. It will be noticed that by this means the coils form one continuous winding through their connection to the commutator; that is, all the coils are in series and form a closed circuit.

The coils A, B and C, being under the influence of the N pole, will have induced in them an electromotive

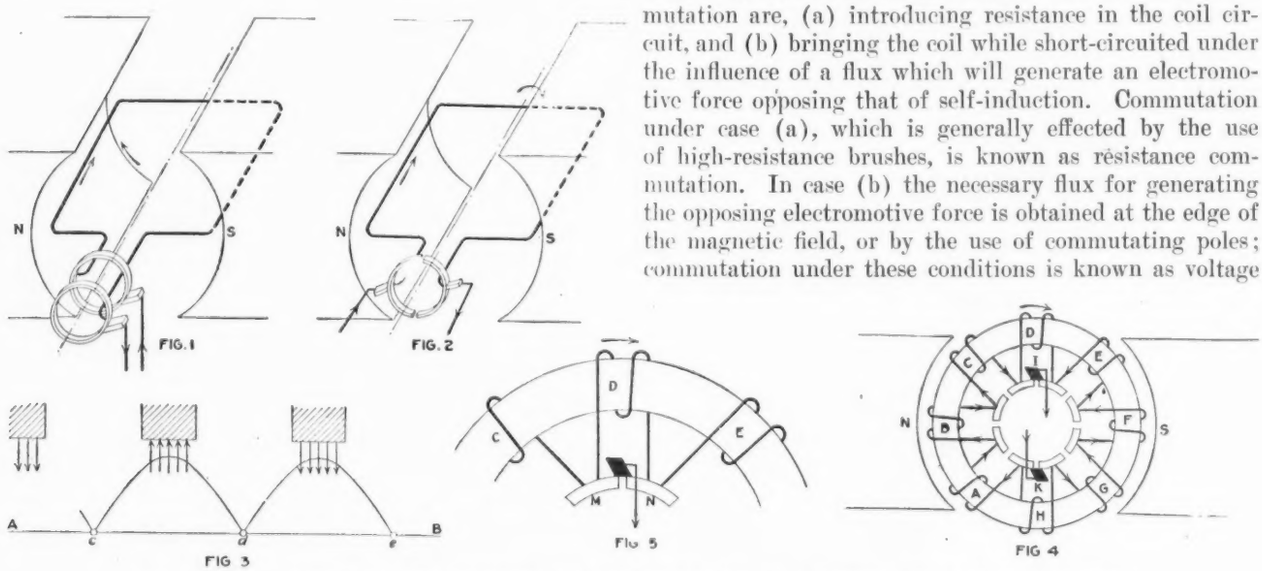
force which will cause a current to flow in the direction indicated by the arrowheads. On the other side of the armature the coils *E*, *F* and *G* are cutting the flux in the opposite direction and will have induced an electromotive force opposite to that of the coils *A*, *B* and *C*. The current in these coils will therefore flow in a direction opposite to that in coils under the *N* pole, but its path from the armature will be seen to be by way of brush *I*, and returning, by brush *K*, the same as for the first set of coils. It is seen that there are two paths in parallel through the armature for the current and that the difference of potential between the brushes *I* and *K* is the sum of the electromotive forces generated in all the coils on one side of the armature.

It will be observed that the coils *D* and *H* are short-circuited by the brushes bridging the gap between the two commutator segments to which they connect, and at this instant they do not form part of the circuit on either side of the armature. The coil *D*, when at the position now occupied by *C*, was carrying current in one direction. When it reaches the position of *E*, it will carry current in the opposite direction. Between these two positions the current in *D* must reverse and rise to the same value as that flowing in the coil *E* on the other side of the brush. Likewise, between the positions *G* and *A*, the current must reverse in coil *H*. This reversal takes place as each coil in succession passes a brush, and constitutes the act of commutation. The position of the brushes between the poles in Fig. 4 is the no-voltage point in the armature; that is, commutation takes place when the coil is out of the influence of either pole. This position is known as the neutral point.

As a coil arrives at the position of short-circuit under the brush, the current which it has been carrying as part of the circuit on one side of the armature must stop and reverse in the very short period that the coil is short-circuited and before it becomes a part of the circuit on the other side of the armature. Thus, if *W* is the thickness of the brush in inches and *S* is the peripheral speed of the commutator in inches per second, the time *T* in which reversal occurs, is $T = \frac{W}{S}$.

From the principles of induction it is known that the reversal of a current in a coil sets up a self-induced electromotive force in a direction that opposes the reversal of the current. The value of this electromotive force depends on the rate at which the change of current takes place and the number of turns in the coil undergoing commutation. Therefore it is of importance that the number of turns be kept as low as possible, one turn per coil being the ideal condition. Owing to the self-induced electromotive force the current in the coil does not sink to zero as soon as the coil is short-circuited under the brush, the time required for reversal depending on the value of the self-induced electromotive force. Therefore, unless this reversal, through some inherent feature of design, takes place in the time the coil is passing under the brush—that is, in time *T*—or can be forced to do so, sparking will result.

It is seldom that commutation of a coil takes place inherently in the time allotted to it; therefore, other means have to be used to insure this action. These aids to commutation are, (a) introducing resistance in the coil circuit, and (b) bringing the coil while short-circuited under the influence of a flux which will generate an electromotive force opposing that of self-induction. Commutation under case (a), which is generally effected by the use of high-resistance brushes, is known as resistance commutation. In case (b) the necessary flux for generating the opposing electromotive force is obtained at the edge of the magnetic field, or by the use of commutating poles; commutation under these conditions is known as voltage commutation. In practice, use is generally made of a combination of these two methods.



FIGS. 1 TO 5. EXPLAIN THE PROCESS OF COMMUTATION IN A DIRECT-CURRENT MACHINE

Resistance in the circuit of the coil undergoing commutation acts in the following manner: Referring to Fig. 5, which is an enlarged view of the short-circuited coils in Fig. 4, it will be seen that current is flowing into the brush from both sides of the armature, through the commutator segments *M* and *N*. These segments and the brush also form part of the circuit of coil *D*. As the armature rotates and the segment *N* passes out from under the brush, the amount of brush surface in contact with *N* becomes less, and the resistance of this contact surface correspondingly increases, thus increasing the resistance in the path of the current from the right-hand side of the armature. At the same time the segment *M* is moving farther under the brush and the resistance between this segment and the brush is decreasing. If the brush is of high-resistance material, such as carbon or graphite, an increasing drop of potential will occur at the contact surface of the brush and segment *N*. The difference of potential between these two will act as an electromotive force to cause the current to take the path through the short-circuited coil and into the brush, in-

stead of through segment N, with the result that reversal of the current will be assisted.

The effect of using brushes of low-resistance material, such as copper, can be readily seen. Even up to the time that the segment N leaves the brush the resistance of this contact area is low compared with that of the coil, consequently, current from the right-hand side of the armature will continue flowing to the brush by way of segment N, rather than through the coil D and segment M. The current in coil D will gradually decrease without reversing up to the time that N leaves the brush and the coil becomes a part of the circuit on the right-hand side of the armature. At this instant the current has to rise to full value in the opposite direction in the coil, and as it is impossible for this action to take place instantaneously, owing to the inductance of the coil, sparking will result.

Resistance commutation, while effective in certain cases, cannot be relied upon of itself to insure absolute freedom from sparking. As stated before, it is almost always employed in conjunction with other means for introducing an electromotive force into the short-circuited coil. This is most conveniently done by taking advantage of the fringe or weak field that exists at the edge of the poles. In order to shift the position of the short-circuited coil from the neutral point midway between the poles and bring it within the influence of this flux, the brushes are moved forward a few segments in the direction of rotation on a generator and backward on a motor. The coil is thus brought within the field of the polepiece, and if this position is carefully chosen, a flux of just the proper strength can be found to set up an electromotive force in the short-circuited coil that will oppose the electromotive force of self-induction. It will be observed that this electromotive force is in the same direction as that in the coils on the side of the armature that the short-circuited coil is approaching, so that when the latter leaves the position of short-circuit, it has had the current in it reversed. Commutation by this method is known as voltage commutation, and the electromotive force generated is known as the commutating electromotive force. It must be remembered that this electromotive force is due to the conductors of the short-circuited coil actually cutting the flux at the edge of the polepieces.

Voltage Required for Commutation

The magnitude of the required commutating voltage will depend on the value of the self-induced electromotive force in the coil, which in turn will depend on the current output of the machine. Therefore it would appear that in order to obtain the required electromotive force for commutation, at various outputs of the same machine, it would be necessary to shift the brushes to bring the short-circuited coil within the proper field, but this is not true except in the case of machines equipped with metal brushes. Where carbon or other high-resistance brushes are used, a fixed brush position is possible over comparatively wide ranges of load, with freedom from sparking. Even in this case, however, at overloads a limit may be reached beyond which sparkless commutation is not possible, owing to the fact that the flux decreases because of armature reaction on the edge of the pole toward which the short-circuited coil is shifted. Just at the time when the need for the maximum flux is greatest, it has decreased to such an extent that it may not generate the required electromotive force for reversing the current in the coil under commutation, and sparking occurs. A further shift of the brushes will not result in the required electromotive force being generated, for this will only cause a further shifting of the field flux. A fixed position of the brushes is therefore found for the ordinary load of the machine, and this will generally result in commutation being satisfactory at other loads within its range.

The fact that it is necessary that the short-circuited coil be located at just the right point in the magnetic field and that this field shifts with varying load has been one of the chief reasons for the use of interpoles. From the foregoing it will be seen that the requirement for a commutating electromotive force is that it be zero at no-load, when no current is to be reversed in the coils under commutation, and increase in proportion to the load. The interpole placed in the commutating position between the main poles and excited by being connected in series with the armature provides more nearly at all loads a commutating electromotive force of proper strength than any other available device.

The interpole consists of a narrow pole covering a portion of the armature circumference approximately equal to the pitch of one or two slots, depending on the design of the machine, and has a winding composed of turns connected in series with the armature. The magnetomotive force set up by the interpole winding must be great enough to provide the commutating flux and in addition offset the magnetomotive force produced by the armature ampere turns. This is understood when it is considered that the current in the armature winding sets up a magnetomotive force in the region between the main poles, which is in a direction opposite to that required to produce the commutating flux. Therefore only the ampere turns on the interpole in excess of those necessary to oppose the armature magnetomotive force are productive of commutating flux. This latter is always proportional to the load of the machine, provided saturation of the interpole section does not take place, and renders commutation independent of the main field flux and its variable features.

The Pantocrat All-Metal Slide Rule

Some years ago George W. Richardson, of 4212 West 24th Place, Chicago, undertook to popularize the slide rule by furnishing, at prices much lower than those at which ordinary rules were sold, rules made of metal with a celluloid facing. Circular openings were provided in which letters corresponding to certain desired settings appeared. The idea met with a ready reception, but the instability of the celluloid coatings interfered with sustained accuracy. Mr. Richardson now announces that he has devised a process whereby he impregnates aluminum with white celluloid, making a material which takes a durable smooth white surface for the ruling and which is not affected, as was the celluloid, by climatic and temperature changes. As all the slides are made from the same die, they are interchangeable, and those of various sorts may be used in any of the rules. Hence the name, "Pantocrat." The 10-in. rule is only 1⅜ in. wide, and $\frac{3}{16}$-in. thick.

The ordinary Mannheim rule is furnished with sine, logarithm and tangent scales, or an extra scale is fur-

nished whereby problems in addition and subtraction may be solved. In another type two polyphase scales are furnished on the slide, one ordinary polyphase CI scale divided decimally, another folded polyphase inverted scale divided not decimally but into halves, quarters, eighths and sixteenths, which permits one to multiply or divide directly mixed numbers of this series not exceeding ten. Another slide has four scales, three of which are folded and are divided to read the log of log in conjunction with the lower, or D, scale. Above these is a C scale and above this on the face and stock of the rule a D scale and CI scale. This arrangement permits the rapid solution of problems involving roots or powers by direct operation and at one setting. Fractional powers or roots are handled with equal facility, and natural or hyperbolic logarithms may also be read.

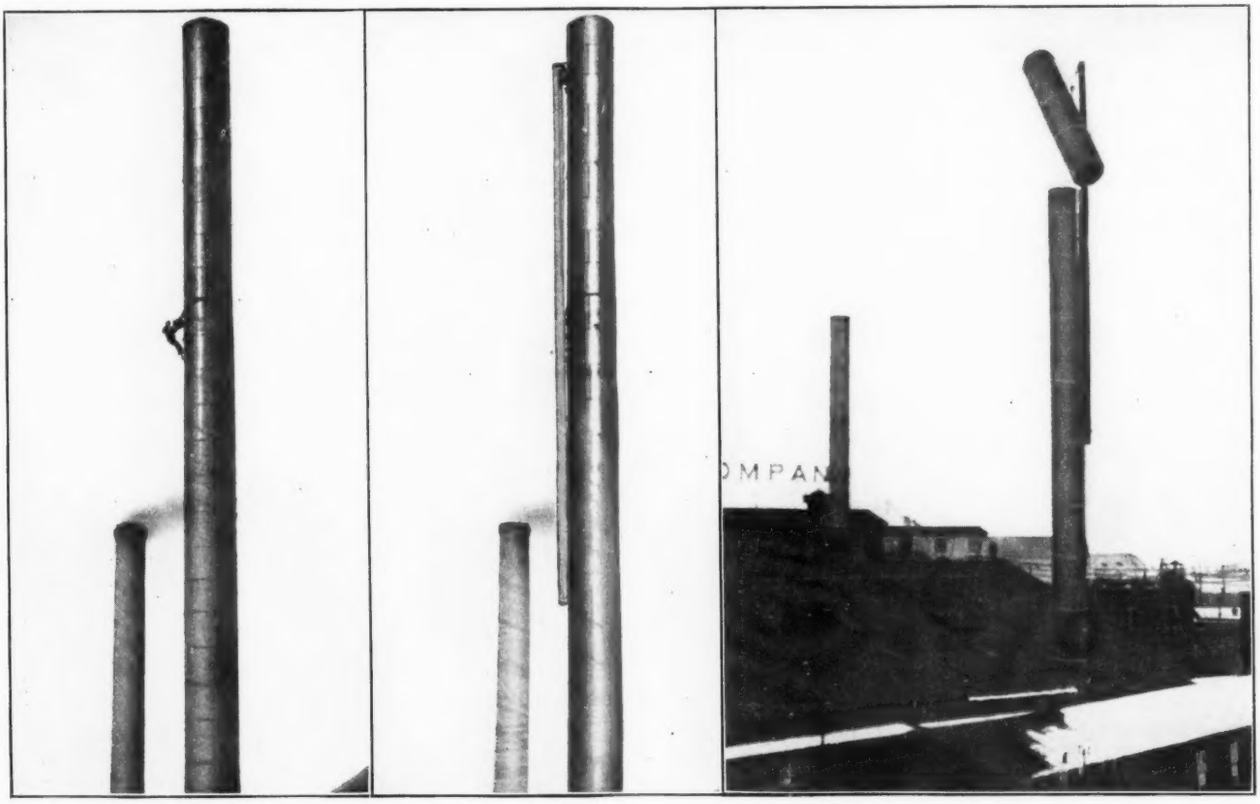
The latest product is a Merchant's slide rule, especially adapted to ordinary mercantile calculations and simplified by leaving off the trigonometrical and other scales not involved in such computations. The two upper scales are the ordinary A and B scales of the Mannheim rule, the first marked "Int." the second, "Days," with the "year" indicated at 365. In the center of the scale is an inverted scale with red figures. The bottom of the slide has a "rate" scale with the index near the center reading to the left up to 3½ and commencing at 3½ at the right and reading to the left back to the index. The lower scale is the same as the two upper. A book is furnished showing how the rule may be used for the simple solution of problems in percentage, taxes, discounts, chain discounts, profit and loss, stock, interest, bank discounts, wages, distribution of accounts, board measure and similar calculations.

Taking Down an Iron Stack

By G. S. Slingerland

An iron stack had to be taken down without being "all mussed up," as it had been sold to see service again in another state. To the average person, taking down an iron stack 150 ft. high and 10 ft. diameter is no small job. It took about a week's time with the aid of a hoisting engine, a lot of cable, blocks and tackle and four or five men, together with a foreman who knew his business and performed his work in an apparently easy and yet systematic way.

The first step was to hoist a gin pole alongside the stack, which measured 16 in. square and 80 ft. long and was born and grew up somewhere in the State of Washington, but finding its way to Buffalo, it is doing all sorts of stunts. To place the gin pole in position a large block was hauled up with a cable and secured to the stack about 100 ft. from the ground. The gin pole was then hoisted until it reached nearly to the top of the stack, being securely fastened at the bottom to the stack and steadied by guy lines from the top, where a large block and fall was fastened with which the top section of the stack was to be lifted off by the help of the hoisting engine.



PROCESS OF REMOVING A STACK IN SECTIONS TO BE USED AGAIN

After all lines had been drawn tight and a strain taken on the top of the stack, the rivets in one seam were cut, from the inside, and after the workman had got out of harm's way, the topmost section was raised clear and then lowered to the ground. By lowering the gin poles and repeating the process, the stack was dismantled in three sections. All very easy when you know how—and have the equipment.

Steam Engine Troubles—Valves

By H. Hamkens

SYNOPSIS—The second article of a series on Corliss engine troubles. Steam and exhaust valves should be drawfiled and scraped to make them tight. Some good and some bad valve designs are shown. Separators are discussed.

The steam and exhaust valves of a Corliss engine require much attention. In buying and installing an engine, it is important to be sure that the valves and the portholes are perfectly round and parallel. If they are not, it will take a long time to get the valves to seat properly, with that smooth, oil-soaked surface which is desired to make them run easily and with the least amount of friction. Some manufacturers grind the valves on special machines with emery or carborundum wheels, but this is a rather dangerous thing to do. Unless the valves are of very close-grained iron and are carefully cleaned after the grinding process, some particles of emery may remain in the pores of the cast iron and quickly grind out the valve seat. Emery, carborundum and all sharp cutting materials are bad things to use on the moving parts of an engine. A good file or scraper is best. The most satisfactory way of making them tight is to bore the portholes round and straight and then drawfile the valves to a good fit. Valves fitted in that way, while perhaps not tight at first, will soon come to a good, lasting bearing. After a few days' run the new valves should be taken out and examined and the high places removed with a smooth file or scraper.

Single-ported Corliss valves are seldom used on new engines, as they require a long travel and give slow steam admission and release, which is accompanied by wiredrawing and loss of economy. Double-ported valves and cylinders are essential for modern Corliss engines. The steam and exhaust passages into the cylinder should be as short, and the flow of the steam and exhaust as direct, as possible without complicated and tortuous bends. Fig. 24 shows a good construction of steam valve. As indicated by the arrows, the change of direction is easy and gradual. The exhaust valve shown in Fig. 25 has the same characteristic features; besides, it is located high up in the counterbore of the cylinder—an important factor in steam consumption, since the clearance is reduced considerably. The arrangements of steam and exhaust valves shown in Figs. 26 and 27 are undesirable, though they are found in many Corliss engines. In Fig. 26 the steam has to pass over the top of the valve, and it enters the cylinder after changing its direction twice, which results in a loss of pressure. Fig. 27 shows the exhaust valve located below the counterbore of the cylinder, with long passages and increased clearance. No high steam economy can be expected from designs like Figs. 26 and 27. Another undesirable steam valve is shown in Fig. 28. It is a so-called double-ported valve with a single steam port in the cylinder. This kind has a large clearance and the steam pressure acting on the valves when closed is much increased, consequently they open hard and the wear is excessive. Besides, the valve-gear parts are likely to give trouble on account of the heavy strain put on them in opening the valve. Trouble may also be expected from the valve in wearing a shoulder on the seat, which after a while will need reboring of the ports and, probably, new valves.

Some exhaust valves are designed as shown in Figs. 29 and 30. Neither can be recommended, since they will

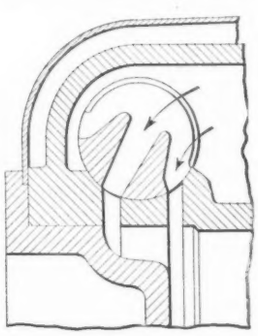


FIG. 24.

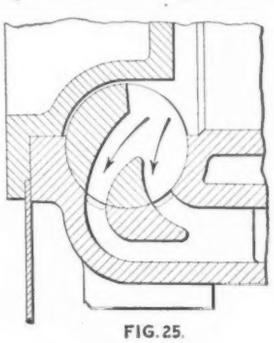


FIG. 25.

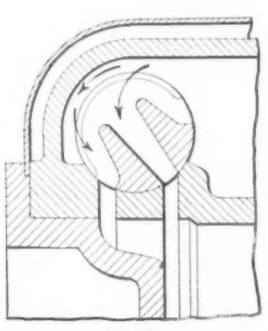


FIG. 26.

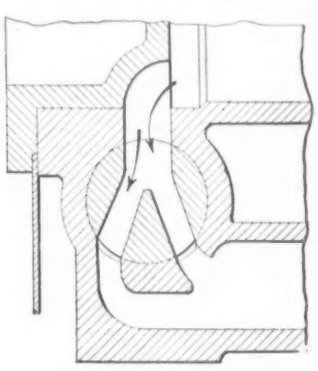


FIG. 27

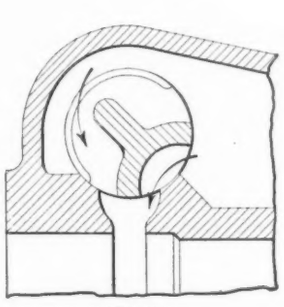


FIG. 28.

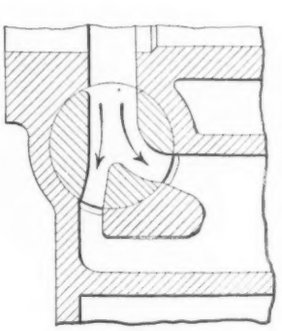


FIG. 29.

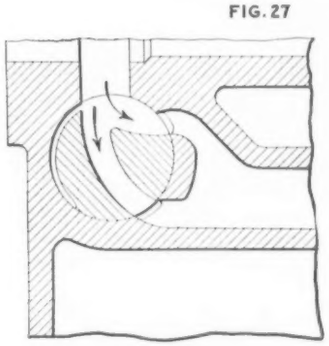


FIG. 30.

SECTIONS OF TYPICAL CORLISS VALVES AND STEAM PASSAGES

eventually leak and give trouble, especially the one shown in Fig. 30. The seat of a Corliss valve should always be downward and not at an angle to the vertical line. Gibs and springs to hold valves onto the seats should be avoided.

Steam valves should be made so that they will lift off their seats and give relief in case the compression should run too high through a disarrangement of the

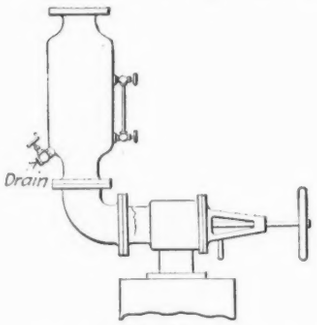


Fig. 31

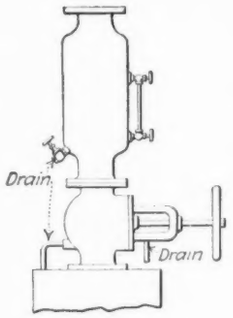


Fig. 32

THE WRONG AND THE RIGHT WAY TO CONNECT UP A SEPARATOR

valve gear or slipping of the eccentric, or if a considerable amount of water should get into the cylinder. What engineers call "a dose of water" is a dangerous thing and must not only be guarded against, but taken care of in the most approved way. The number of cracked cylinders and blown-out cylinder heads caused by neglect to provide relief valves is legion. Therefore a relief valve of ample size should be located at each end of the cylinder in as low a position as possible. The relief valves and the lifting steam valves are the only "cure" that we know of to counteract the destructive effect that will result from water in any considerable quantity being left in the cylinder after the exhaust valves have closed. To prevent water from reaching the cylinder at all, it is of the greatest importance that the steam piping should be well covered, either with asbestos or 85 per cent. magnesia, to reduce the condensation to a minimum, and also to have it well drained. There should be no possibility of water accumulating gradually in pockets and then discharging into the cylinder all at once. Even steam separators may become dangerous and a source of trouble, if the water is allowed to collect in them. Water cannot be compressed to any appreciable extent, and an engine is not designed like a pump. The piston speed is much too high and the area of the exhaust valves is not large enough to discharge a great quantity of water at anything like the usual piston speeds. The velocity in well-designed pumps rarely exceeds 5 ft. per sec., or 300 ft. per min., which is about one-half of the ordinary piston speed of steam engines. The valve area of pumps is equal to or even larger than the piston area, while the exhaust ports of a steam cylinder are only about 15 per cent. of the piston area, so it can be easily understood why an engine is wrecked if any considerable amount of water is to be discharged from the cylinder, even with the exhaust valves wide open. The relief valves can take care of only a small quantity of water at the end of the stroke, when the exhaust valve is closed and the piston travel reduced, and so can the steam valves by lifting off their seat.

The combination of steam separator and angle throttle valve shown in Fig. 31 is not to be recommended. In this arrangement there is a bending strain in the neck of the valve owing to the weight of the separator and steam piping; water-hammer may also give trouble, and breaks in the neck are not infrequent. The separator and steam pipe should be placed directly over a globe valve, as in Fig. 32. Drains should be provided as shown.

[*Receivers is the subject of the next article*]

Design of Oil-Ring Bearings

By William Knight*

SYNOPSIS—In which the main points to be considered in designing oil-ring bearings are discussed. These include the question of oil grooves, their width and depth, the direction of rotation of the shaft, the bearing metals used and the advantages of one over another, also the proportion of oil rings and the number required for each bearing. The prevention of leakage of oil from bearings is also dealt with.

A correct design of bearings is one of the main points to be considered when designing a machine. If they are not suitable for the particular service that the machine is intended to perform, trouble may be expected during its operation, and no matter how well designed all other parts are, the final result will be a failure.

The function of bearings is to withstand the load imposed on them and to allow for a free rotation of the shaft with the least frictional loss, without undue wear, vibration or oil leakage; this being particularly important in electrical machines and high-speed rotors in general in which vibrations are always objectionable.

It is the aim of this article to describe the main points to be considered when designing a bearing in order to obtain the desired results, chief of which is the supplying of a sufficient amount of oil all along the journal, so as to maintain an unbroken film.

In order to point out the importance of supplying the oil at the right place a résumé is given of the investigations on the points of maximum and minimum pressure in the oil film, made by Tower, Moore and others.

Tower's experiments (*Proc.* Inst. Mech. Eng., 1885) were made on a half-bearing 4 in. diameter and 6 in. long; chord of arc of contact 3.3 in. Nine holes were drilled in three longitudinal rows, one at the bottom and the other two rows spaced 1 in. from the central one, and located one at the side where the journal entered and the other where the journal left the bearing.

The distance between holes in the same row was 1 in., each row starting at approximately the center of the bearing and extending toward one end. Each hole was connected to a pressure gage, the average bearing pressure (total load divided by the projected area of contact) was

*Engineering department, Crocker-Wheeler Co.

311 lb. per sq.in. and the oil (bath lubrication) was kept at a constant temperature of 90 deg. F. The pressure at the three holes nearest to the middle of the bearing was found to be a maximum at the bottom hole, or twice the calculated average pressure. At the hole on the leaving side of the bearing the pressure was 60 per cent. more than the average, and at the entering side it was 19 per cent. more than the average. Gage readings at the other holes showed in every row a slight gradual drop in pressure from the middle plane toward the end of the bearing, the pressure at the end holes being an average 87 per cent. of the pressures at the middle of the bearing. All these pressures were found to vary directly as the load on the bearing and to be independent of the journal speed (between 150 and 20 ft. per min.) This, however, does not agree with Reynold's theory, which shows that the distribution of the pressure in the film varies with the speed.

Beside having points of maximum and minimum pressure in the oil film, H. F. Moore in 1898 (*American Machinist*, Sept. 10, 1903) proved with some tests on a small bearing that at some points of a complete circular bearing, the pressure in the film may even drop below the atmospheric pressure and suck oil into the bearing from a level 6 in. below.

Pressure on Oil Film

This point was first proved mathematically by Reynold (*Phil. Trans.* Royal Soc. London, 1886). When the bearing is first started from rest, the point of contact between the journal and journal box moves back in a direction opposite to the rotation a small angle, following the law of dry friction. As soon as the oil starts lubricating the rubbing surfaces, the point of maximum approach is gradually shifted about 90 deg. in the direction of the rotation. By increasing the load beyond a certain limit or by decreasing the speed of rotation (load and speed being inversely related in Reynold's equation), the point of nearest approach of the journal and bearing surfaces is shifted back from the horizontal plane (load on the bearing assumed to be acting vertically downward) to 40 deg. from the vertical on the off side of the journal, and the thickness of the oil film at that point is, according to Reynold, equal to one-half the difference of the radii of the journal and journal box.

When the point of nearest approach on the off side is 40 deg. from the vertical, the pressure in the oil film is everywhere greater than the atmospheric pressure.

By further decreasing the speed or by increasing the load, the point of nearest approach is shifted farther back in a direction opposite to the rotation toward the bottom of the box, the thickness of the film becomes less, and the pressure reaches a maximum value at a point slightly before the point of maximum approach and becomes negative at a point slightly beyond that point, until finally, if the pressure is still further increased a certain amount or else the speed decreased, the oil film breaks at the point of negative pressure.

By increasing the speed, the extreme position of the point of maximum approach in a bearing with the load acting vertically downward is on the horizontal plane at 90 deg. from the vertical in the direction of the rotation. Up to a certain speed the points at which the pressure is respectively a maximum and zero are located very near to the point of maximum approach—the former just before, the latter just beyond. By further increasing the speed, these two points begin to diverge from the horizontal plane for a corresponding increase in thickness of the oil film at the point of nearest approach, with a tendency at very high speed to become diametrically opposite to each other in a vertical plane when the journal is located exactly in the center of the journal box (a position which is not reached in practice).

These investigations on the location of the point of maximum and minimum pressure with reference to the plane in which the load on the bearing acts lead us to the determination of the most desirable point at which the bearing must be supplied with oil.

Evidently the oil must be supplied in that region where the pressure is a minimum, unless forced lubrication is used. In order to find that point, it is necessary to first determine the direction of the resultant of the forces acting on the bearing (gravitation only, or else gravitation and whatever external force is applied to the shaft). The point of minimum pressure in a bearing subject to a moderate pressure and rotating at a speed not too low is in a plane located, in the average case, at about 120 deg. from the plane where the resultant of the forces on the bearing is acting, measured in the direction of the rotation; and the point of maximum pressure is located at about 60 deg. back of it (or 60 deg. from the plane of the resultant, measured in the direction of the rotation), provided that a perfect oil film is maintained all around the shaft.

Providing Oil Grooves

If the direction of the resultant was always known, it would be easy to provide oil grooves in the most convenient places, which would allow the oil to be admitted at the right points; but in some cases, as for instance, in electrical motors, the direction of the pull on the shaft due to the belt or to a gear cannot always be anticipated and the bearing must be designed to provide the best lubrication possible under the conditions.

The bearing surface where the pressure is a maximum should be as free as possible from oil grooves, and the first oil-ring lubricated bearing, used in 1860 by Benjamin Hick & Son, Bolton, England, was a failure because the oil grooves were cut in the lower half of the bearing instead of in the upper half, where they would have been effective.

Oil grooves should be designed, first, to conduct the oil to the center of the bearing from where it spreads out toward the ends; second, to provide on the bearing surface a reservoir of oil which will always maintain a generous supply of oil throughout the length of the bearing; and third, to catch the oil toward the ends and return it to the previously mentioned reservoir, thus preventing, as much as possible, the escape of the oil from the journal box.

The direction of the rotation should also be considered in designing the oil grooves for a given bearing, because if they are not cut in the proper direction, they will not assist in carrying the oil from the center supply to the ends as intended. In electrical motors designed for right- and left-hand rotation it is necessary to strike a happy medium in providing grooves, which are a necessary evil in bearing design, and they must be cut symmetrically for both directions of rotation. Too many grooves will unnecessarily reduce the bearing surface and will assist

in breaking the film. Improper grooving will have the same effect.

Having provided for the distribution of the oil over the surface of the bearing where the pressure is a minimum, provision must be made so that it is not wiped off and is allowed to reach the surfaces more heavily loaded. This is provided for in bearings loaded on the bottom half with some secondary grooves (chamfers) cut lengthwise at the split of the bottom half of the journal box and extending to within a short distance from the ends, or by counterboring the journal box at the split in such a way as to allow the oil to collect there and to be drawn into the lower half between the journal and journal box over the length of the bearing.

In general, no oil grooves should be provided in the lower half of a bearing with the load acting on the bottom half, except two shallow grooves extending around the lower half of the journal box (in large sizes only) near the ends, in which the oil that is squeezed out can collect and be again carried into the chamfer or the counterbored recess at the split on the other side.

Grooves and chamfers it is generally assumed should be cut smoothly and have no sharp edges which might scrape off the lubricant. That sharp edges are detrimental to the maintenance of a perfect film is rather questionable, and I know of an engineer who has proved with numerous tests the fallacy of the opinion generally held by manufacturers and users of journal boxes in regard to this point.

Width and Depth of Oil Grooves

The width and depth of oil grooves is another matter to be considered. They should be wide and deep enough to insure a sufficient supply of oil to be distributed all over the bearing surface.

The size of the bearings governs the size of the oil grooves, but in most cases from $\frac{1}{8}$ to $\frac{1}{4}$ in. width and $\frac{1}{16}$ to $\frac{5}{32}$ in. depth are sufficient. Besides accomplishing this main function, the oil grooves will be useful if metal rubbing should occur with a consequent rubbing off of the bearing metal. In that case the abraded metal will collect into the grooves and fill them in before seizing occurs, and the machine can be frequently stopped in time to avoid serious damage to the bearing if the heating is noticed in time.

When a bearing shows a tendency to heat, it is well to inspect the oil grooves, because they may have become partly obstructed by the babbitt or other material and may prevent the flow of the oil.

J. C. K. Balfry, in a paper read before the Rugby Engineering Society, states that many firms in England are designing high-speed bearings without any grooves at all and they believe it to be sufficient to supply oil along the whole length of the bearing at about the split. The theory of the high-speed bearing developed by Reynold shows that grooves in such a bearing do not serve any purpose, and it may even be argued that they are actually detrimental.

It is a maxim that similar metals must not be used for journal and bearing, except in the case of hardened steel and cast iron. This is probably so because they are of the same hardness, and it is conceivable that crystals of the same hardness have some sort of affinity and combine together under favorable conditions by alloying or welding, more easily than crystals of different hardness.

Low-carbon steel journals are likely to give trouble where high-carbon steel journals would give none, and this fact also shows that in the case of low-carbon steel a greater affinity is likely to be found between the journal and some crystals which may become detached from the bearing metal.

Gray unchilled cast iron, owing to the presence of graphite and to its porosity, makes excellent bearing surfaces when working in contact with steel or wrought iron, and also against itself. It does not seize readily and the only objection to its use as a bearing metal is its brittleness.

Bearing Metals

For electrical machines two classes of bearing metals are generally used—bronzes and white metals. The fundamental difference between bronze and white "antifrictional alloys" is to be found in their microscopic structure. In bronzes plastic crystals of copper are embedded in a hard eutectic, and if, by reason of defective lubrication, metal contact should occur at any spot between the journal and the bearing surface, particles of copper will separate from the mass and become welded or alloyed to the journal (especially if this is made of soft steel), forming a rough surface which will attract at every revolution fresh particles of copper until serious heating and damage result.

White metals instead are composed of hard grains having a low coefficient of friction (which for this reason do not easily cut the journal) embedded in a plastic alloy.

Bronzes are alloys composed chiefly of copper and tin. The presence of tin confers strength to the alloy, which up to about 13 per cent. of tin is a tough homogeneous solid solution of tin in copper. If the percentage of tin is increased beyond this limit, the alloy becomes hard and brittle and increases in hardness and brittleness in proportion to the increased percentage of tin.

Composition of Bearing Metals

Phosphor bronze is an alloy of copper, tin and phosphorus. When used in the proportion of from 84.5 to 91.0 per cent. of copper and 0.37 to 0.85 per cent. of phosphorus (see *Journal* Inst. Metals, page 164, 1909), a network of hard copper-phosphide embedded in a soft matrix of copper-tin alloy is formed and a metal particularly suitable for bearing service will result, with a low coefficient of friction. According to Philip such an alloy is particularly suitable for high-speed machines.

The addition of lead (provided that the metal is well stirred up before pouring and is not poured at a too high temperature, so as to insure a good diffusion of the lead throughout the molten mass) greatly increases the plasticity and reduces the wear. An alloy of 79.7 per cent. of copper, 10 per cent. of tin, 9.5 per cent. of lead and 0.8 per cent. of phosphorus was proved by Dudley's experiments (*Journal* Franklin Inst., 1892, pp. 81-93 and 161-72) to be a metal sufficiently strong, plastic, of good wearing properties and of a finely granular structure so as to make it particularly suitable for bearing metal. The addition of lead to bronze alloys was the first step toward the adoption of plastic white-metal linings in rigid-bronze, cast-iron or steel journal boxes.

Lead is a cheap metal, but is too plastic to be used alone and must be hardened by adding antimony and tin. Babbitt metals were eventually introduced and are now

largely used. The original proportions used by Babbitt as given in Alford's book, "Bearings and Their Lubrication," were:

Tin	=	89.3 or 83.3 or	89.1
Copper	=	3.6 or 8.3 or	3.7
Antimony	=	7.1 or 8.4 or	7.2

White metal and babbitt alloys are particularly suited for the lining of bearing shells owing to their plastic nature, which enables them to mold themselves to the form of the journal, thus providing a larger bearing surface with a more even distribution of the load. High spots can be more easily scraped down, and a smooth and polished bearing surface is obtained. Bearing surfaces of white metal and babbitt are also more economical than bronze, because of the comparatively low cost at which a new lining can be provided in a worn-down or otherwise damaged journal box.

White metals are alloys of tin, antimony, lead, copper and zinc. Other elements, like iron, bismuth, aluminum and graphite, enter in the composition of the alloy, and as for bronzes, the different proportions into which the constituent parts are alloyed together will provide a bearing metal more or less suitable for a given service. For instance, for bearings subjected to heavy pressure or severe shocks a hard mixture is required; a soft one would be rubbed off in flakes. An alloy of 80 parts of lead, 12 parts of tin, 7.5 parts of antimony and 0.5 part of copper is a good bearing metal for heavy loads. An alloy composed of 80 parts of lead and 20 parts of tin is particularly suited for high-speed bearings and moderate loads. A Brinell hardness of 23.5 for lead-base and 30 for tin-base babbitts have been found to give excellent results.

Cause of Hot Bearings

As already pointed out, white-metal alloys are made of hard grains embedded in a plastic matrix, and as for bronzes, a coarse size of grains is to be avoided in order to prevent damage to the shaft.

Babbitt linings must be provided with a sufficient number of anchoring buttons to secure a firm connection with the cast-iron shell. This is an important point in bearing construction, and the cause of many hot bearings can be traced back to a lack of contact between the bearing metal and the outside shell.

This lack of contact may be produced by the shrinking of the babbitt when poured in the journal box, due to the difference in the coefficient of expansion and temperature of the babbitt and the bearing shell (the coefficient of expansion of the babbitt being two or three times larger than that of cast iron). When secured with buttons to the outer shell, the buttons will prevent the babbitt from becoming loose, but for this very reason shrinkage stresses are produced in the babbitt, which under shocks and heavy loads will develop cracks that will become gradually impregnated with oil. The presence of an oil film between the babbitt and the outside shell at those places where a good contact is not obtained, owing to the very low heat conductivity of the oil, will prevent the dissipation of the heat from the babbitt to the shell and from this to the atmosphere, thus increasing the temperature of the babbitt.

To correct these troubles as much as possible, an intelligent choice should be made of the most suitable alloy to be used for a given service, and proper consideration should be given to its physical characteristics (tensile and compressive strengths, elongation, reduction of area and hardness). A sufficient number of buttons should be provided for, and great care should be taken when pouring the babbitt that no dross or any other foreign substance is present in the liquid mass of metal, and overheating of it should be carefully avoided.

The practice of one of the largest electrical machine manufacturing concerns is to pour hard babbitt at an average temperature of 775 deg. F and soft babbitt at an average of 920 deg. F., allowing not more than 115 deg. F. variation both ways. Overheating is more serious in soft babbitt than in hard. The shell should be heated to between 210 and 300 deg. F. before pouring in the babbitt, and babbitted bearings should not be jarred during the solidification of the metal.

In order to obtain as good a contact as possible between the lining and the shell in a journal box, the bearing metal lining must be carefully peened. An ideal metal would be one expanding during the process of solidification instead of contracting, and in this class are antimony alloys.

Bearing Clearance

Between the journal and the journal box a sufficient clearance must be provided to insure a good film lubrication and to take care of the expansion of the babbitt. The coefficient of expansion of the babbitt being two or three times larger than that of either cast iron or steel and the babbitt being prevented by the shell from expanding outwardly, the expansion will have to take place inwardly, and if a sufficient clearance is not provided for, the oil film may be broken and binding may occur. A clearance of 0.001 in. per in. of bore plus 0.002 in. will be about right for electrical motors and generators, provided that the deflection of the shaft inside of the journal box, due to the weight of the rotor, the belt pull or an overhanging load, is less than the clearance between the journal and the journal box.

Large outfits are generally equipped with spherically seated journal boxes, which will insure a better alignment of the shaft and a more uniform thickness of the oil film due to a better equalization of the load over the entire length of the bearing. With self-aligning bearings, a clearance of 0.001 in. per in. of bore plus 0.002 in. will do.

Oil-Ring Bearings

With oil-ring lubricated bearings another point to be considered is the design of the oil ring. In order to have efficient lubrication, these rings must be properly designed and constructed. It is generally assumed that an oil ring cannot be expected to distribute oil over a width of bearing of more than 8 in. (or 4 in. at each side of it), so that the maximum length of bearing with which a single ring at the center could be used is 8 in.; for bearings between 16 and 8 in. long two rings should be used located at a distance from the ends equal to about one-quarter the bearing length; for a bearing 24 in. long, three rings should be used, and so on. As a matter of fact, however, the writer knows of bearings 27 in. long where two rings only are sufficient to bring up an amount of oil sufficient to insure good lubrication.

Good proportions for oil rings are obtained by making them about twice the diameter of the journal and providing an oil level in the oil well at a distance from the shaft about equal to half the diameter of the journal.

All corners and sharp edges in the ring should be avoided, and in split rings there should be a good contact between the two halves at the joint, so that the oil cannot squirt out through the split. Rings should be made perfectly round and well-balanced in order to prevent "dancing" of the ring on the shaft. When dancing occurs, the ring stops and no oil is brought up to the journal. This gradually increases the friction between ring and journal until the ring, because of the increased friction, starts rotating again, brings up some oil, then stops and starts dancing again.

Preventing Oil Throwing

Owing to defective working of the oil rings, oil is sometimes thrown against the bearing cap and, owing to capillary action, creeps out through the split. Oil leakage at the split is to be particularly avoided with electrical machines on account of the destructive effect of oil upon the insulation of the coils and commutator. This can be prevented with a groove at the split of the pedestal which will drain the oil reaching there into the oil well again without allowing it to leak out, or else an oil guard may be cast in the bearing cap, extending below the split and inclosing the oil ring.

Another place where oil troubles may be anticipated and efficiently prevented with a good design is at the shoulder of the journal. At this place, owing to the end play of the shaft, the oil is squeezed out between the shoulder on the shaft and the end of the journal box. To prevent the oil from being thrown out of the bearing, a groove is cut in the journal box either into the bearing surface adjacent to the thrust shoulder or else outside of it, opposite to the oil sling, with drain holes at the bottom which will allow the return of the oil into the oil well If the latter construction is used, a small clearance should be provided between the journal and the journal box at the oil sling, such as to insure a sort of sealing effect, and the oil sling on the shaft should be made wide enough to prevent the lip of the journal box from ever clearing the shaft and thus destroying the desired sealing effect, owing to the end play.

Adjacent to the oil sling an oil groove is provided on the shaft, so that whatever oil may happen to leak out of the journal box over the oil sling is driven back again by centrifugal force without being allowed to get out of the bearing. In some designs, adjacent to this groove another oil sling similar to the one adjacent to the thrust shoulder is cut into the shaft. Opposite to these two slings a second oil groove could be made in the journal box, similar to that previously described and accomplishing the same function. In this way the oil, being squeezed out between the thrust shoulder and journal box, is forced to go in a first chamber in the journal box having drain holes at the bottom.

If the amount of oil being squeezed out into that chamber is too much, the excess which does not flow through the drain holes will leak out and will fall in the groove on the shaft, whence it will be thrown back again, a part going back to the first chamber and another part filling the outer chamber in the journal box.

The same effect could be obtained by having a thrower mounted on the shaft near the thrust shoulder surrounded by the bearing housing, which would catch the oil thrown out of the bearing in a direction parallel to the axis of the shaft.

Dust should not be allowed in the bearing. This may be provided for by using a felt or leather washer fastened to the bearing housing and wiping on the shaft.

Another important point to be considered is a rigid construction of the pedestals or shields supporting the bearings and a good alignment of the shaft. A springy support might cause trouble, and a bad alignment of the shaft will offset all the best care taken in properly designing bearings.

Mono CO_2, SO_2, and O Recorders

Although there may not be a great demand in the factory power plant for an instrument that will record percentages of carbon dioxide (CO_2), carbon monoxide (CO), sulphur dioxide (SO_2) and oxygen (O) in the exit gases of boiler furnaces, there has been a want for

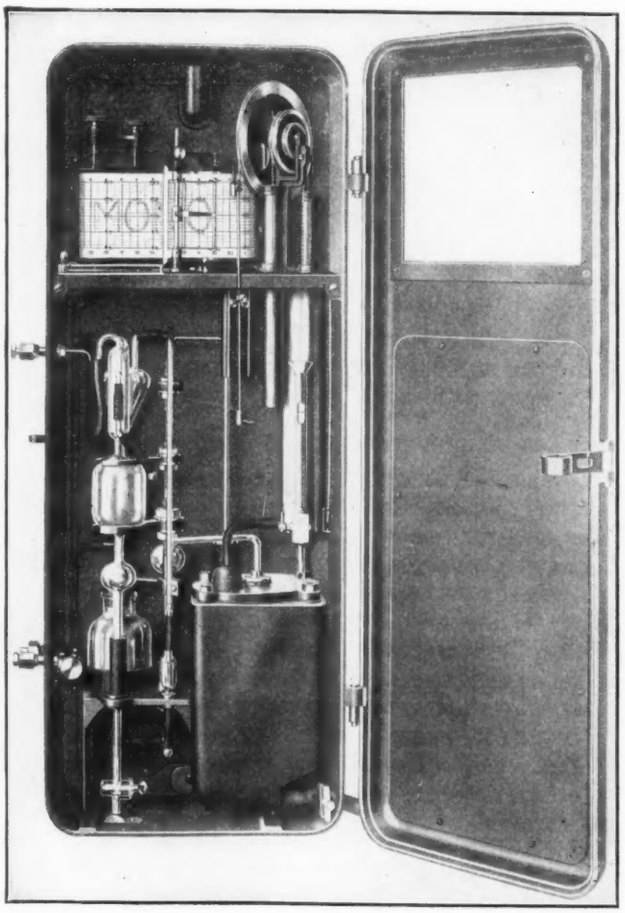


FIG. 1. MONO CO_2 RECORDER

just such an instrument in the various branches of the chemical, paper and pulp mills as well as in cement and lime industries. Various types of carbon-dioxide (CO_2) recorders are on the market, but oxygen recorders, SO_2 recorders and monoxide recorders are now offered to the various industrial establishments for the first time, in so far as is known. Of the apparatus described herewith, Fig. 1 is the Mono CO_2 recorder.

The absorption apparatus is mounted in the lower part of the cast-iron cabinet. The pressure medium (water or air) by which the apparatus is driven passes through a regulating valve and the pipe *B*, Fig. 2, into the flask

C, which contains mercury which is forced up into the pipes, *D*, *E*, *F* and *G*.

Pipe *D* is connected at the top with the burette in which the volume of the gas is ascertained. Pipe *E* is in communication with the outside air, and the upper part of pipe *F* passes into an expansion of the pipe *G*.

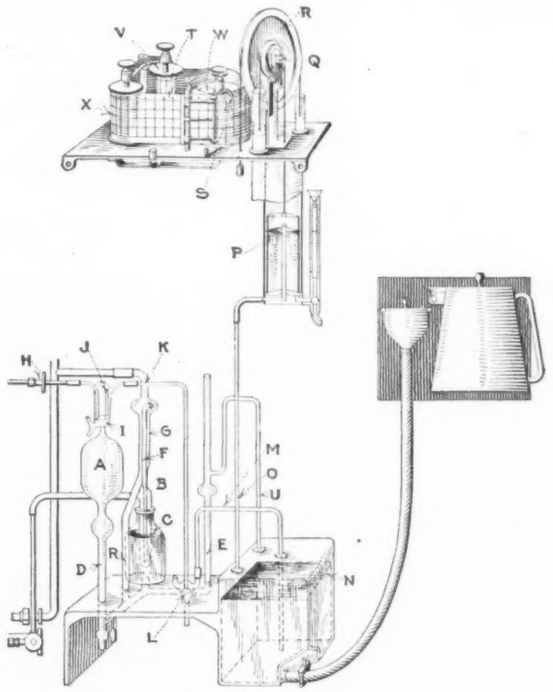


FIG. 2. PIPING OF MONO CO_2 RECORDER

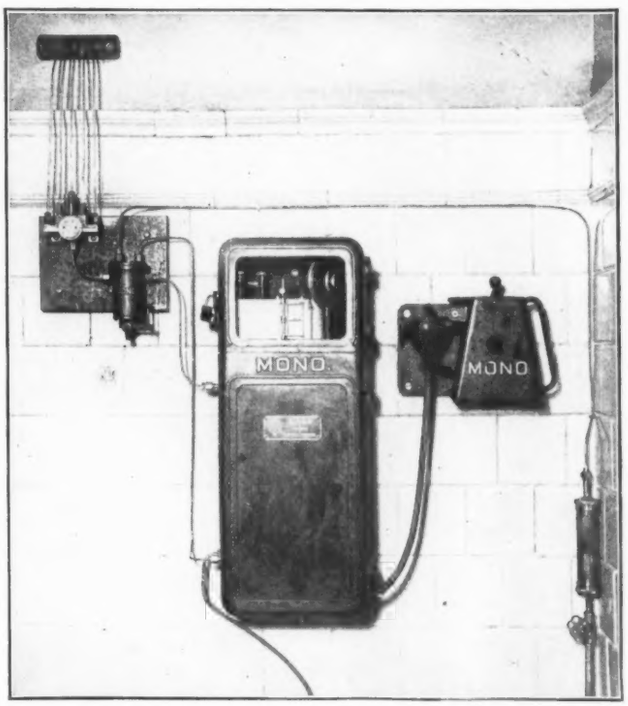


FIG. 3. ONE RECORDER CONNECTED TO FIVE BOILERS FOR CO_2 GAS SAMPLES

The lower part of the pipe *F* passes into the bottom of a recipient *B* and is thus only indirectly in connection with the mercury in the flask *C*. The position of this inner recipient in the flask is such that the mercury in the pipe *F* will always rise higher than that in pipes *E* or *G*.

As the pressure is increased, the mercury is finally forced out of the recipient *B* through the pipe *F* and runs down into the pipe *G* or *E*. As soon as the pipe *F* is empty of mercury, the pressure medium comes in contact with the outside air and the excess pressure in the flask *C* disappears. The mercury in the pipes *D* and *E* then runs back into the flask *C*, filling it until some finally runs into the recipient *B*. By this means the lower outlet of the pipe *F* is closed. The pressure medium is then no longer in contact with the outside air, the pressure rises again in the flask *C*, and the process described is repeated. In this way an alternately rising and falling movement of the mercury is brought about, which movement is utilized as described in the following paragraphs.

When the mercury sinks, as already described, the gas to be analyzed is drawn in from the chimney through the filter and the tubing to the absorption apparatus.

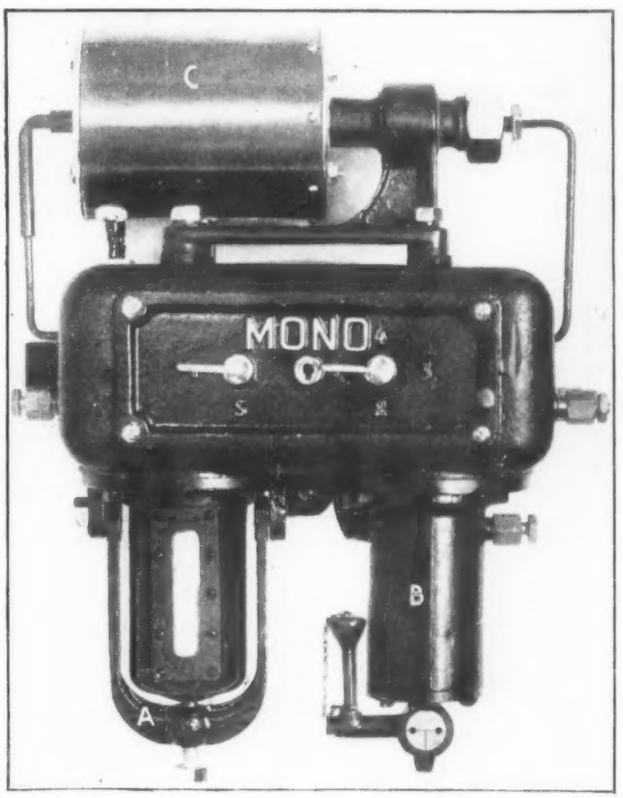


FIG. 4. CO_2 RECORDER AUXILIARY APPARATUS

It then goes through the coupling *H*, the mercury seal *I* and the pipe *J*, and passes into the burette *A*, where it is measured, the pressure and temperature always being constant. When the mercury rises, the gas from the burette is pressed through the pipe *K*, the mercury seal *L* and pipe *M* into the recipient *N*, which is filled with absorption liquid. The carbonic or sulphuric acid in the gas is here absorbed, the remainder of the gas being pressed through the pipe *O* up into the gasometer *P*, which is shut off from the outside air by water.

When the gas enters, the gasometer rises and turns the wheel *Q*, which in its turn finally acts on the wheel *R*. On this wheel and connected with it by means of a metal chain hangs the pen *S*, which draws the curve on

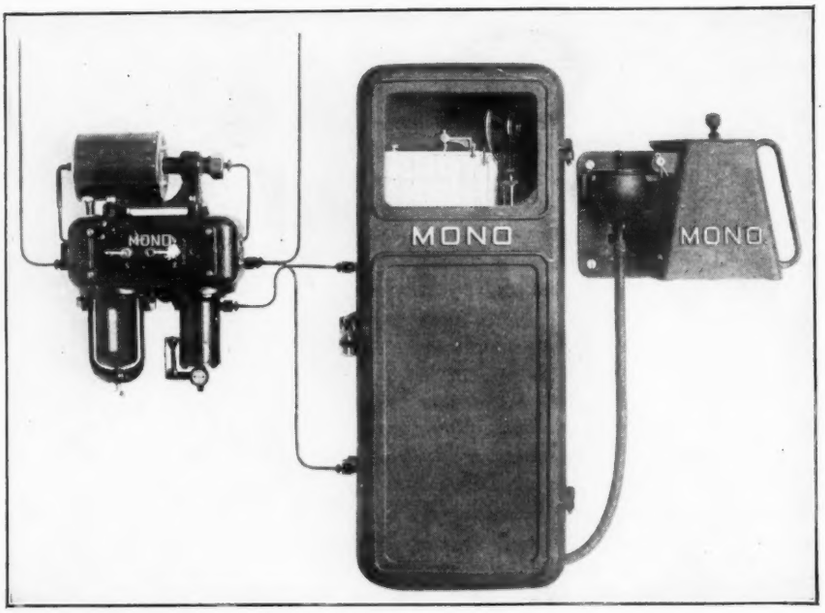


FIG. 5. MONOXIDE RECORDER

the diagram paper T. The lower extremity of the line produced indicates the percentage of gas absorbed.

When gas is drawn into the burette, the pipe U comes in contact with the outside air, the gasometer then sinks back to its original position and the apparatus makes a new analysis and a new registration. The chart is drawn from the roller V over the drum W by means of a clock in the latter and afterward automatically rolls up on the roller X or is cut off at will.

There is no water or other injector used with this instrument to draw the gases from their respective sources. The pipe line employed for connecting the instrument with the furnaces consists of $\frac{1}{16}$-in. copper or brass tubing. In order to prevent clogging of the $\frac{1}{16}$-in. pipe with soot, a simple cotton-waste filter is placed at the intake of each flue, and owing to the intermittent pull of a mercury column within the recorder, just the volume of gas required for analysis is drawn into the instrument. If clogging of the pipe should occur (in cases where a very high volatile coal is used), there will be no faulty recording, as the machine will instantly stop and the operator will immediately know the cause of the trouble.

The 50 ft. of piping as used with this instrument contains about the same volume of gas as that of the burette in the instrument. Thus, if the burette fills and empties itself only once—that is, after the first cycle—the correct sample as existing in the flue is being analyzed. Inasmuch as the instrument can be operated so as to record more than 60 analyses an hour, it will be seen that the lag does not exceed one minute when 50 ft. of pipe is used. As this heavy pull could not be accomplished by a column of water, mercury is used.

This understanding of the Mono CO_2 recorder will assist in making clear the operation of the Mono SO_2 recorder inasmuch as the same instrument is being used, only that the parts subject to corrosion from the sulphuric-acid fumes are built of sulphur-dioxide resisting materials. A caustic-potash solution of the specific gravity of 1.27 Baumé is used the same as in the CO_2 recorder in all cases where the gas to be analyzed does not show any measurable quantities of CO_2 and SO_3, which would also be absorbed by the caustic-potash solution and thus give an erroneous result. In cases where SO_2, together with volumes of SO_3 and CO_2, are present, caustic potash as an absorbent cannot be used, and for such cases a special recorder is made, using an iodine solution which will absorb only the SO_2, thus giving a correct analysis of this gas. Owing to the use of such iodine solution, the gasometer P, Fig. 2, is built of glass. These instruments are mostly used in sulphite mills of the paper and pulp industries, as well as in plants where sulphuric acids are manufactured. Fig. 3 shows the method employed to connect one recorder with five boilers, using a five-way valve. The Mono monoxide recorder, Fig. 5, can be used not only as such, but also as a CO_2 recorder. It will be seen that the CO_2 recorder is provided (at the left) with an auxiliary apparatus, Fig. 4. The gas drawn in by the recorder enters this apparatus and passes through an absorbent for CO_2 in A. Thus the CO_2 is eliminated, leaving only carbon monoxide and hydrocarbons. This residue passes into the electric oven on the top of the auxiliary apparatus C, where it again is burnt to CO_2 and after passing a temperature regulator and moisture absorber B, it enters the CO_2 recorder and there the CO_2 is absorbed, as already described. Thus the per-

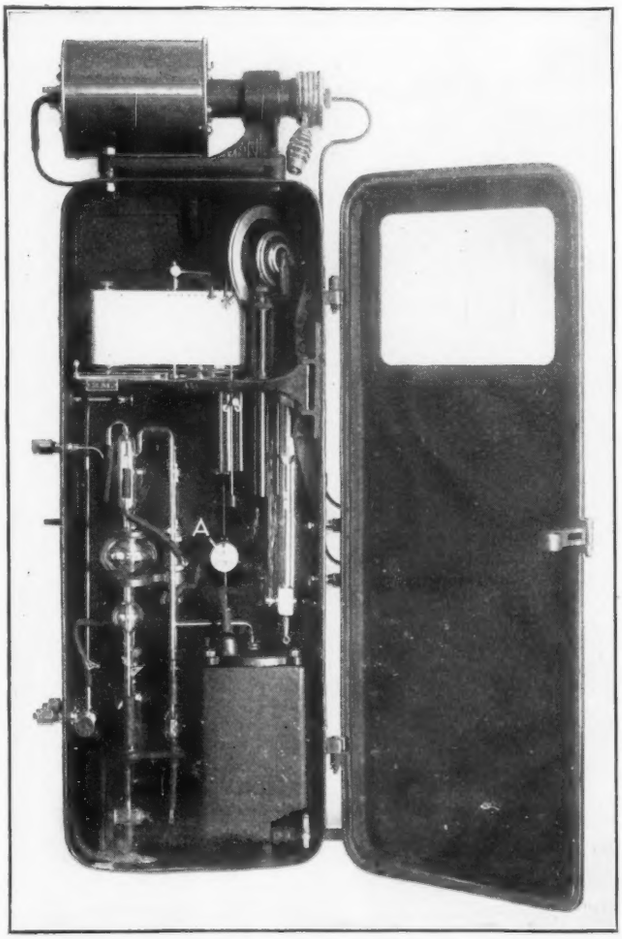


FIG. 6. OXYGEN-CO_2 RECORDER

centage shown on the chart represents the percentage of CO contained in the gas.

The Mono oxygen-CO_2 recorder is shown in Fig. 6. This, too, is a combination and can be used for recording both gases CO_2 and O. This is accomplished by the simple throwing of a valve *A*. When the valve is in one position, the gas passes through a solution of caustic potash and the machine operates as a CO_2 recorder. When the valve is in the other position, it cuts in the electric oven on top of the machine, simultaneously closing the inlet to the caustic-potash chamber and sending the gas into the oven, where, under a high temperature combined with an oxygen-absorbing medium, the oxygen in the gas is absorbed and the residue is measured again in the gasometer and recorded on the chart.

These instruments, including the oxygen recorder, can make over sixty analyses per hour without impairing their accuracy.

The machines are sold in the United States by F. D. Harger, Park Row Building, New York City.

Safety Devices for Elevators

By Jacob Gintz, Jr.

SYNOPSIS—A description is given of the construction and function of the potential switch and safety devices, as used on an electrical elevator.

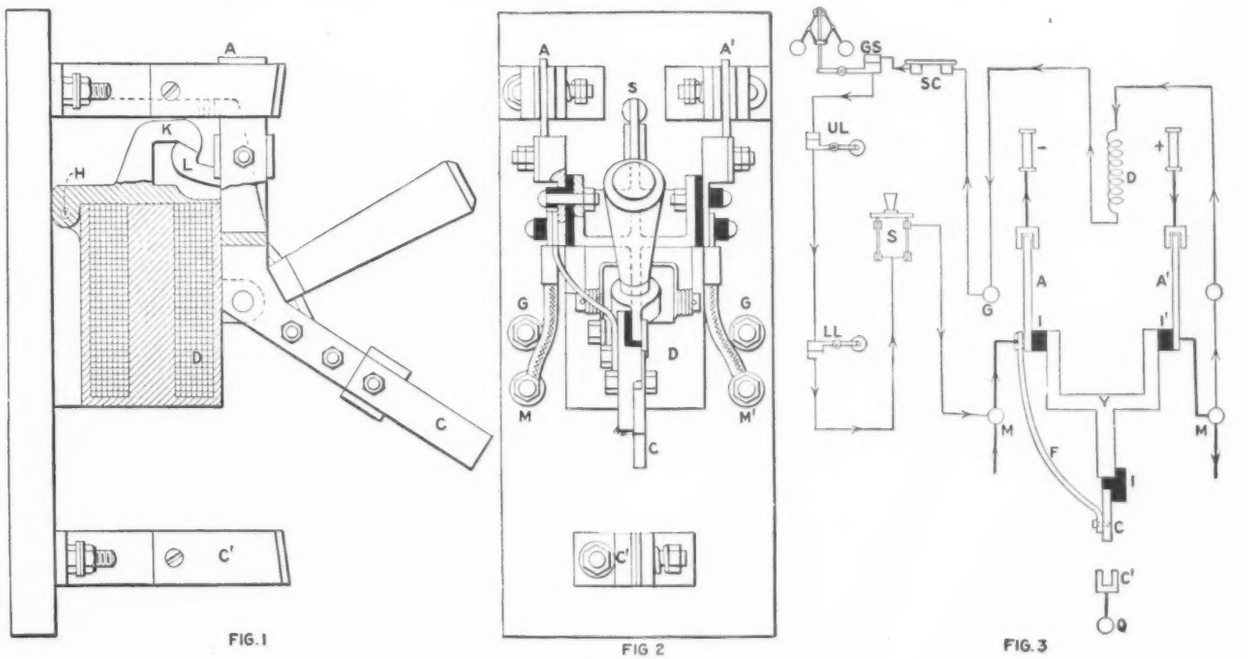
The potential switch as used in connection with electric-elevator machinery is constructed somewhat like a circuit-breaker, but instead of opening the circuit when excessive current is flowing, it opens it when the voltage falls below a certain value or if the safety devices or their circuit is open by accident or otherwise.

The switch has three blades, *A*, *A'* and *C*, and is provided with a no-voltage release coil *D*, as indicated in Figs. 1, 2 and 3. This coil is connected to terminals *G* and is in series with the upper and lower shaft-limit switches, the governor switch, the safety switch in the car and the slack-cable switch, as shown in Fig. 3. If any of these safety devices are opened, there will be no current flowing through the potential-switch coil, therefore the switch will open and stop the machine.

The coil *D*, Fig. 1, is wound with many turns of fine wire and is fitted over an iron core. The entire coil is incased in iron with the core in the center forming one pole and the rim of the case the other pole. The cap is hinged at the point *H* and has a hook *K*. This hook holds the catch *L*, which is attached to the movable parts of the switch. When the circuit through this coil is opened by any of the safety devices being interrupted, the cap is released and the catch *L* is no longer held, allowing the switch to be forced open by the spring *S*, Fig. 2.

The safety devices are a very important part of an electric elevator and are known as follows: Safety switch in the car, slack-cable switch, governor switch, upper- and lower-limit switches and the no-voltage release coil on the potential switch. On some of the modern types of electric elevators gate and door contacts are installed which are also classed as safety devices. The gate or door contacts are in series with the magnet coils on the reversing switches and the car-switch circuits. If either one of these two contacts is not closed, it will be impossible to operate the car.

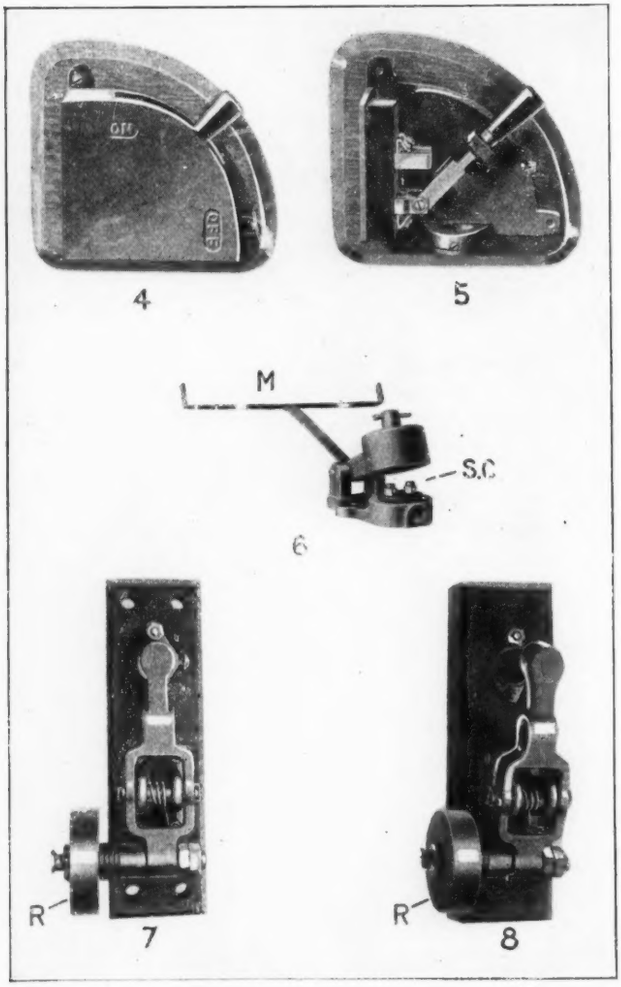
The safety switch, Figs. 4 and 5, is a small double-pole knife-switch with its two blades joined at *S*, Fig. 3, and is used in case the car gets beyond the control of the opera-



FIGS. 1 TO 3. POTENTIAL SWITCH AND CIRCUIT THROUGH SAFETY DEVICES

tor. By opening this switch the potential switch will be opened, which will cut the power out of the motor and set the brakes, thus stopping the machine.

The slack-cable switch shown in Fig. 6 and at *SC*, Fig. 3, is set on the bedplate of the machine underneath the drum; should the car jam in the shaft so that the cables become slack on the drum, they will fall on the arm *M*, open the switch *SC* and shut down the motor by opening the potential switch. The slack-cable switch is shown in an open position in Fig. 6.



FIGS. 4 TO 8. ELECTRIC ELEVATOR SAFETY DEVICES

The governor switch *GS*, Fig. 3, is placed in the overhead work and is operated by the flyball governor on the car safety cable, Fig. 9. If the car exceeds a certain speed, the governor will open the switch and at the same time set the safety shoes on the car, which will grip the rail and bring the car to a stop. The opening of this switch opens the potential switch, which in turn will stop the motor.

The upper- and lower-limit switches, Figs. 7 and 8, are a protection against the car being sent into the overhead work or striking the bottom of the pit. The upper-limit switch *UL* is set a little above the upper landing and the lower limit *LL*, Fig. 3, is set a little below the lower landing, the exact distance depending upon the amount of overhead or pit room. These switches are opened by cams, which are fastened to the top and bottom of the car, striking the rolls *R*, Figs. 7 and 8. The opening of either switch will open the potential switch. The no-voltage release coil will hold the potential switch closed as long as the foregoing safety devices are closed, since they are all in series with one another and in series with the no-voltage release coil.

The door contacts are placed in the jamb of the door on each floor and the gate contacts on the car. They are in series with one another and with the magnet coils on the reversing switches, which control the armature circuits of the motor, consequently, until the floor door and car gate are closed, the car cannot start.

The following is an explanation of the wiring diagram for the various safety devices shown in Fig. 3: *A* and *A'* are the upper blades of the potential switch to which the line wires are connected from the fuses; *Y* is the metal yoke of the switch and is insulated from the blades at *I*; *C* is the lower blade, which is connected to one of the upper blades *A* by a flexible wire *F*; *C'* is the clip for the blade *C*; the terminals *M* lead to the motor and starting devices; terminal *Q* is connected to a section of the starting resistance through which the armature is short-circuited when the potential switch is open and acts as a dynamic break.

By tracing through the connections it will be seen that the current flows from the positive fuse to *A'*, through the no-voltage release coil, the slack-cable switch, the governor switch, the upper- and lower-limit switches and the safety switch in the car, back to the negative fuse, as shown by the arrowheads.

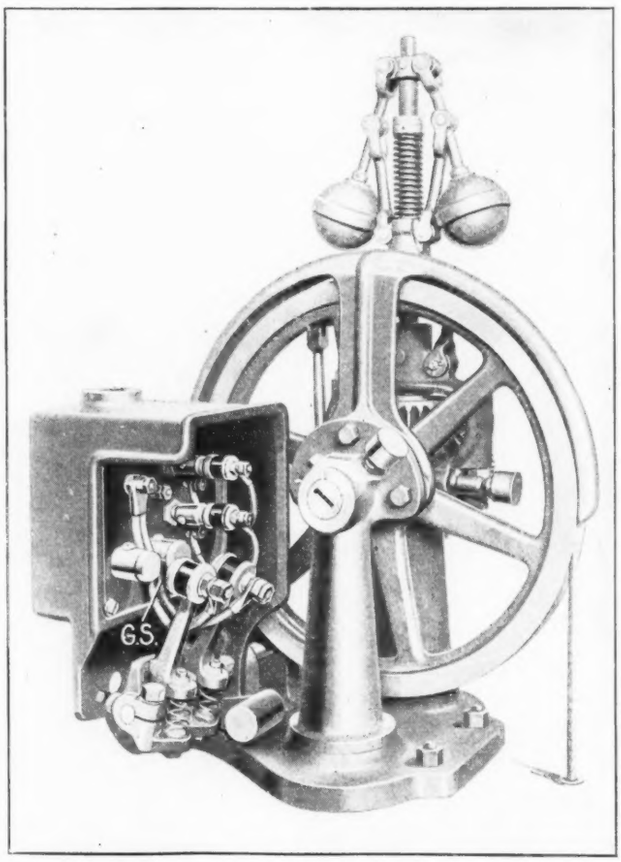


FIG. 9. ELEVATOR GOVERNOR AND GOVERNOR SWITCH

Surface Condensers are allowed from 1½ to 2½ sq.ft. of tube surface per indicated horsepower or, better, 1 sq.ft. tube surface for 8 to 12 lb. of steam to be condensed per hour.

Steam Engineer's License Examination—I

By H. F. Gauss

SYNOPSIS—The first of a series of articles outlining the requirements for a steam engineer's license examination. The general subjects to be covered are steam boilers, pumps, engines, elevators and auxiliaries. Much information that will be useful in the examination will be given, and there will be references telling where other data may be found.

There is no necessity for an operating engineer being able to manipulate complex mathematical equations or to discuss elaborate theories relating to purely physical or chemical phenomena. On the other hand, the young engineer makes a fatal mistake when he scorns theory and concludes that "practical experience," as he terms it, is all that is necessary. A line between what should be called theory and what should be called practical knowledge is becoming hard to draw, but it is not the purpose of this article to define either of the two extremes.

The amount of theory involved in the series of articles to follow ought not to be beyond the grasp of the operating engineer, though some of it may not be considered essential for safe power-plant operation. The articles were suggested by an examination recently given in St. Louis, Mo., for a steam engineer's license, and this account should be helpful to others in preparing for similar examinations. To follow the articles and construct the models suggested will require some time and pains, but no great mathematical ability.

Work of the St. Louis Board

It may be said here that the examining board in St. Louis is composed of three competent engineers, all holding responsible positions in the steam-engineering field. Their examinations are considered thorough and practical; and while the theoretical reasons for various practices will be gone into more thoroughly in this review than the board requires of a candidate, yet the general scheme of examination will be followed throughout.

On application for examination each candidate is furnished with a brief printed outline of the field the test is to cover and also a copy of the city ordinances relating to the operation of steam appliances. With this material as a guide and backed by the proper practical experience, the applicant should have no serious trouble in passing a satisfactory examination. The general subjects covered are steam boilers, pumps, engines, elevators, auxiliaries.

Under the subject of steam boilers is included a brief discussion of the principal types of boilers in use—the fire-tube or return-tubular, the water-tube, the vertical and the marine. The candidate must be able to show by neat sketches that he thoroughly understands the construction and setting of these boilers. It is a good idea in preparing for such an examination to secure catalogs and descriptive matter from various boiler manufacturers and to practice sketching the settings illustrated therein.

The Heine Safety Boiler Co. publishes a book called "Helios," in which is a typical section through a standard hand-fired Heine boiler. The Babcock & Wilcox Co. publishes "Steam," in which a typical section through a hand-fired B. & W. boiler is shown. The Kewanee Boiler Co. publishes a catalog showing standard settings for horizontal return-tubular boilers, and S. F. Freeman & Sons Manufacturing Co., Racine, Wis., fully describes marine-type boilers in its catalog. All these and other catalogs can be obtained for the asking by anyone directly interested.

An essential qualification for a successful candidate is an understanding of the design and construction of riveted joints, a knowledge of the size, pitch and arrangement of rivets in the various types, including single-, double- and triple-riveted lap and double-, triple- and quadruple-riveted double-strap butt joints. The applicant must be able to sketch a typical joint of each class, analyze the possible methods of failure, determine the pitch, edge distance, etc., and in fact design the complete joint. On pages 63 to 69 of the Kewanee catalog are given sketches and details of various joints, which should assist materially in learning to make the above-mentioned sketches.

Before being able to design a riveted joint, the designer must fully understand what is meant by ultimate stress

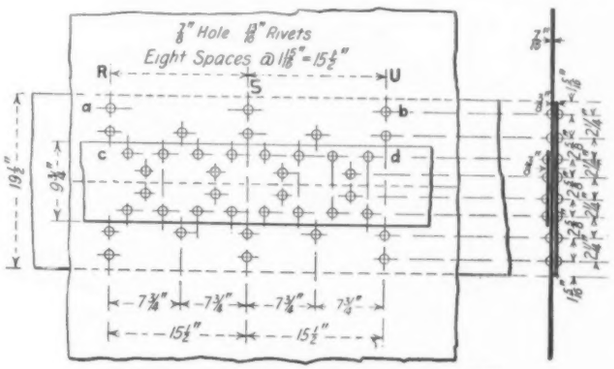


FIG. 1. STANDARD TYPE OF JOINT

and safe working stress. In the first place there are three distinct kinds of stress—tensile, compressive and shearing. Each is the stress, or load, that will cause a bar of a given material of 1-sq.in. cross-section to fail in the manner and from the stress indicated. For instance, the ultimate tensile stress per square inch is the load in pounds pulling on a bar of 1-sq.in. cross-section that will just cause it to break or fail; similarly for the shearing and crushing stresses, respectively. The safe stress is the load in pounds per square inch which the material can safely withstand; that is, the load that, when applied to the bar in any of the three manners indicated, will not cause it to take a permanent set. Knowing the stress that will cause a bar of 1-sq.in. section to fail, it is easy to compute the total load that will break a bar of any number of square inches in section. Likewise, knowing the safe load for a bar 1 sq.in., it is easy to determine the safe load for a bar of greater or less section.

The following may be found convenient for fixing in mind the design of riveted joints and shells and solving problems that may arise in an examination, knowing first:

1. Of what are the plates to be made (steel or iron)?
2. Of what are the rivets to be made (steel or iron)?
3. What are the allowable ultimate stresses in tension, shear and compression?

Let

f_t = Ultimate strength in tension, lb. per sq.in.;
f_s = Ultimate strength in shear, lb. per sq.in.;
f_c = Ultimate strength in compression, lb. per sq.in.

The following values for these quantities may be assumed when the actual strength is not known:

f_t = 55,000 lb. for steel;
f_t = 50,000 lb. for iron.

The allowable stress on stay-bolts shall not in any case exceed 7,500 lb. per sq.in.

f_s = 44,000 lb. for steel;
f_s = 38,000 lb. for iron;
f_c = 95,000 lb. for mild steel;
f_c = 90,000 lb. for iron.

4. What is to be the factor of safety; that is, by what number is the ultimate stress to be divided to obtain the safe working stress?

Condition of Boiler	Type of Joint	Factor of Safety[1]
New	Lap riveted	5.0
New	Butt (double strap)	4.5
Second-hand	Lap riveted	6.0
Second-hand	Butt (double strap)	5.5

5. What shall be the efficiency of the joint?

This, with the diameter of the shell, determines the type of joint necessary. The lap joint is rapidly going out of favor, owing to difficulties in proper inspection as well as to the eccentric strains that exist in the metal of a joint of this kind. However, for low-pressure work and in shells less than 42 in. diameter the lap joint still is used. The following table gives the efficiencies obtainable with various joints:

Type of Joint	Efficiency, Per Cent.
Single-riveted lap	40 to 57
Double-riveted lap	63 to 74
Triple-riveted lap	72 to 77
Single-riveted double-strap butt	40 to 57
Double-riveted double-strap butt	72 to 82
Triple-riveted double-strap butt	84 to 87
Quadruple-riveted double-strap butt	90 to 94

Having decided on the allowable stresses, the factor of safety and the efficiency of the joint, the following principles are to be borne in mind:

Experiments show that a rivet in double shear is about twice as strong as in single shear. In Kent's "Mechanical Engineers' Pocket-Book" the value taken for a rivet in double shear is only 1¾ times as strong as one in single shear, but authorities do not agree on this point. The A. S. M. E. Code considers a rivet in double shear twice as strong as one in single shear, and this value will here be used. Thus, suppose there are in a double-strap butt joint three rivets in single shear and eight rivets in double shear; that is, three rivets pass through the shell and one cover strap and eight rivets pass through the shell and both cover straps. This is equivalent to having $3 + (2 \times 8) = 19$ rivets in single shear. Assuming that, after driving, the rivet completely fills the hole, it is permissible to use the diameter of the hole in computing the crushing and shearing strength of the joint.

There are three principal ways in which a joint may fail. They are: First, "tension alone"—that is, the plate may tear along the outer row of rivets; second, "shear alone"—that is, all the rivets in the joint may shear off; third, "compression alone"—that is, the plate may crush before each rivet. The joint may fail in many combinations of the single methods mentioned. For example, the plate may tear along an inner row and all the rivets in the outer rows may shear off. Space will not permit an analysis of all the possible ways in which a joint might fail, but the following concrete example will serve to illustrate how the foregoing quantities and principles bear on the design of a joint.

Design a shell 60 in. diameter by 16 ft. long to withstand a gage pressure of 160 lb. per sq.in., the shell and rivets to be of mild steel. The ultimate allowable stresses in tension, shear and compression are: f_t = 60,000 lb. per sq.in.; f_s = 38,000 lb. per sq.in.; f_c = 90,000 lb. per sq.in. The factor of safety is 5. The efficiency of the joint is 94 per cent., which means a quadruple-riveted double-strap butt joint. One rivet in double shear equals two rivets in single shear. Should all these conditions and quantities not be given, they would have to be assumed as outlined in the preceding paragraphs. The following symbols will be used:

t = Thickness of plate in inches;
t' = Thickness of butt strap;
d = Diameter of rivet hole;
h = Pitch of rivets in outside row;
n_1 = Rivets in single shear;
n_2 = Rivets in double shear;
$n = n_1 + n_2$ = Equivalent rivets in single shear;
P = Gage pressure in pounds per square inch;
D = Diameter of shell in inches;
F = Factor of safety;
E = Efficiency of joint.

It may be stated here that E is the lowest ultimate resistance of the joint to failure, divided by the resistance of the solid plate. In general, the efficiency may be determined by the equation $E = \frac{h - d}{h}$. However, as a check, the different manners of failure should be computed and divided by the strength of the solid plate to make sure that $\frac{h - d}{h}$ is the lowest efficiency.

First, the thickness of the plate may be determined from the well-known formula

$$t = \frac{PDF}{2f_tE} \qquad (1)$$

$$t = \frac{160 \times 60 \times 5}{2 \times 60{,}000 \times 0.94} = 0.4375 = \tfrac{7}{16}+$$

The shell, therefore, will be made $\frac{7}{16}$ in. thick and the cover plates ⅜ in. thick.

Fig. 1 shows the arrangement of a standard quadruple-riveted double-strap butt joint. It will be noticed that the arrangement of the rivets repeats itself in successive sections; for instance, section R to S is identical with section S to U, etc. Hence, it becomes necessary to compute the strength of a single section only, the length of this section being the pitch of the outer row of rivets. Let

R_t = Ultimate resistance of joint to tearing along outer row ab;
R_s = Ultimate resistance of joint to failure by all the rivets shearing off;

[1] The reader is advised to compare this with the A. S. M. E. Boiler Code.

R_{st} = Ultimate resistance of the joint to failure by tearing along row *cd* and shearing the rivets in the outer rows;

R_c = Ultimate resistance of the joint to failure by the crushing of the plate before each rivet.

A properly designed joint should be equally strong to resist failure in any of the possible ways. With this type of joint it is found that if $R_t = R_s = R_{st}$ the joint will be sufficiently strong to resist failure by the other methods.

$$R_t = (h - d)tf_t;$$

$$R_s = (n_1 + 2n_2)\,0.7854d^2f_s;$$

$$R_s = (2 \times 8 + 3)\,0.7854d^2f_s = 14.92d^2f_s;$$

$$R_{st} = (h - 4d)tf_t + 3 \times 0.7854d^2f_s$$

$$= htf_t - 4tdf_t + 2.36d^2f_s;$$

$$R_c = n_2dtf_c + n_1dt'f_c = df_c(n_2t + n_1t').$$

Letting $R_t = R_{st}$ and solving for d gives

$$htf_t - dtf_t = htf_t - 4dtf_t + 2.36d^2f_s$$

$$3dtf_t = 2.36d^2f_s$$

$$d = \frac{1.27tf_t}{f_s} \qquad (2)$$

Substituting the quantities of the present problem in equation 2 gives

$$d = \frac{1.27 \times \frac{7}{16} \times 60{,}000}{38{,}000} = 0.877 = \frac{7}{8}\ in.$$

Hence, use a $\frac{13}{16}$-in. rivet.

Setting $R_t = R_s$ and solving for h gives

$$htf_t - dtf_t = 14.92d^2f_s$$

$$h = \frac{14.92 \times d^2f_s + dtf_t}{tf_t} = \frac{14.92d^2f_s}{tf_t} + d$$

Again substituting the quantities of the present problem gives

$$h = \frac{14.92 \times (\frac{7}{8})^2 \times 38{,}000}{\frac{7}{16} \times 60{,}000} + \frac{7}{8} = 16.54\ in.$$

Call $h = 16\frac{1}{2}$ in.

Hence, the pitch of the outer row of rivets is $16\frac{1}{2}$ in.; the pitch of the second row is $8\frac{1}{4}$ in.; the pitch of the third and fourth rows is $4\frac{1}{8}$ in.

A general rule for determining the edge distance is e = from $1\frac{1}{2}$ to 2 times the diameter of the rivet.

The distance between rows should be sufficient to allow ample clearance for dies and dolly and can always be found in a set of standard specifications, such as issued by the Hartford Steam Boiler Inspection and Insurance Co.

After having designed a joint, it is always best to check back in order to be sure that slight deviation from computed dimensions do not materially affect the strength of the joint.

In general, it may be said that for quadruple- and triple-riveted double-strap butt joints d is determined by equating R_t and R_{st}. In other joints d may be determined by equating R_s and R_c. In all cases h may be determined by equating R_t to R_s.

However, it must be borne in mind that, while results obtained by the use of these theoretically correct formulas are safe as far as the strength of the joint is concerned, a second factor must be considered, which may necessitate a deviation from results so obtained. This second factor is determined by mechanical considerations, the chief of which is the calking of the joint to make it steam-tight. The pitch must not be so great that the joint cannot be calked. It is for this reason that the practice of experienced manufacturers should be studied before attempting to design a new joint, and perhaps the methods above outlined are more valuable as a guide to investigating the strength of a given joint than for designing new ones. Braces and stays must be so spaced that the area supported by each is not greater than that which, multiplied by the pressure per square inch, will result in a force stressing the stay beyond the allowable safe limit. The allowable stress varies in different cities, but is seldom in excess of 7,500 lb. per sq.in. In designing diagonal stays the pressure normal to the surface stayed should be resolved into two components, one parallel to the stay and one parallel to the surface stayed. The former is the load on the stay. Thus, the load on the stay in Fig. 2 is

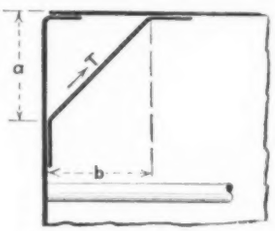


FIG. 2. TYPICAL STAY

$$T = P\frac{L}{b}$$

where

P = Area supported multiplied by the pressure per square inch;

L = Length of stay = $\sqrt{a^2 + b^2}$;

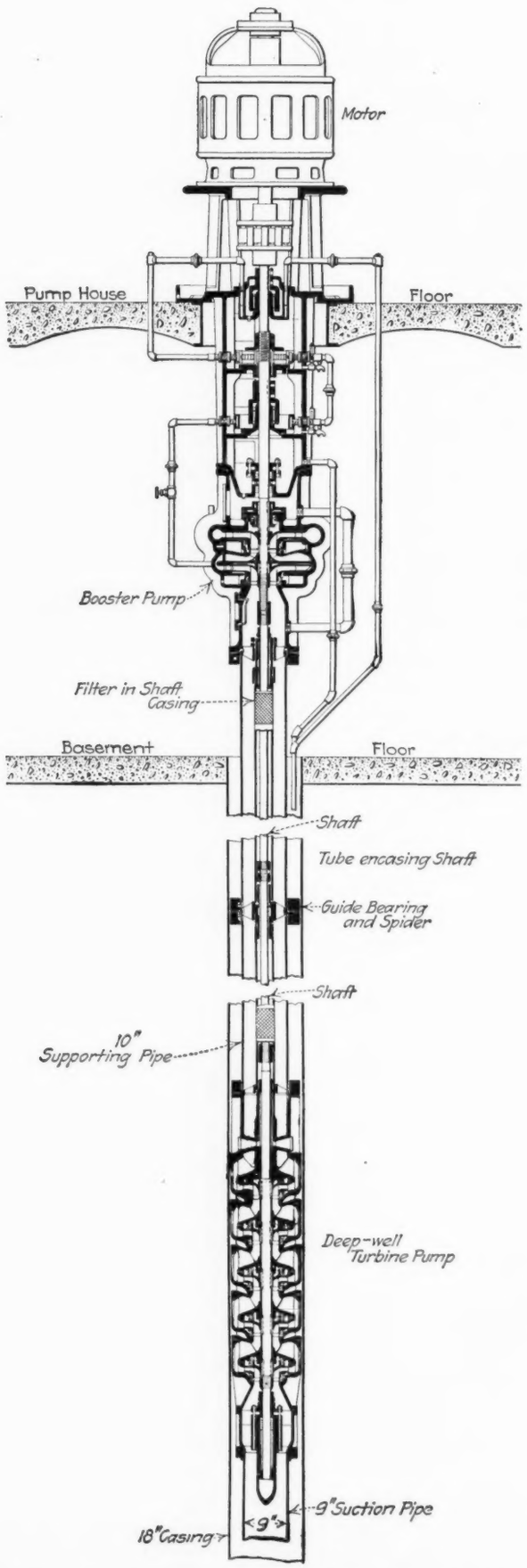
a and b = Distances shown in Fig. 2.

With these points clearly in mind there should be no difficulty in investigating any given shell to determine its safe working pressure, or in designing a new boiler for a given pressure. It should be borne in mind that the factor of safety is separate and distinct from the efficiency of the joint.

Well-Unit Water-Supply at Aurora, Ill.

To provide a necessary increase in the capacity of the water-works of Aurora, Ill., says *Engineering News*, the city has adopted the well-unit supply system. This consists of a number of deep wells distributed throughout the city, each operated by independent pumping units that deliver the water directly into the mains.

Well No. 7 was completed in 1914. The first 300 ft. has 18-in. casing, the balance being in rock, 15 in. inside diameter, to a depth of 2,262 ft. An "American" pump was bought to have a capacity of 1,000 gal. per min. from an elevation 100 ft. below the surface. This unit was to be directly connected to a vertical 150-hp. motor having an automatic starter complete with no-voltage release and overload relays arranged for 50 per cent. full-load start. This installation is herewith illustrated in cross-section showing the construction of the combined deep-well and booster pump. These units are provided with water-cooled vertical roller guide-bearings in the upper frame, and water-cooled roller thrust bearing. The booster and turbine proper both have bronze inclosed impellers, removable bronze diffusers and removable bronze wearing rings.



DEEP-WELL PUMPING UNIT WITH TURBINE AND BOOSTER AT AURORA, ILL.

To increase the pressure up to 105 lb. in the mains for fire purposes, this vertical deep-well unit will work in series with an "American" single-stage double-suction horizontal centrifugal pump directly connected to a 40-hp. slip-ring motor running at 1,800 r.p.m.

The pumps were installed and when tested the well would supply only 640 gal. per min., the water being lowered to 125 ft. from the surface, which was 25 ft. lower than the contract called for. The city therefore decided to lower the pump another 100 ft. The pump was again tested by the builders, and tests were also made by D. H. Maury, consulting engineer, Chicago. The results of the tests are summarized in tabular form and attention is called to the fact that the supply of water did not equal the capacity of the pump.

	Builder's Test	—D. H. Maury's Tests—	
Discharge, gal. per min.....	930	896.0	981.0
Total head, ft.....	424	426.6	410.6
Speed, r.p.m.....	1,145	1,155.0	1,155.0
Pump efficiency, %.....	74½	70.4	70.1
Combined efficiency, %.....	....	62.7	62.8

The method of lubricating the long vertical line shaft that drives the lower pump is this: The line shafting is surrounded or incased in a sectional tube. The sections are fastened together by the line-shaft bearing, and the whole shaft casing is put under tension and steadied by spiders at the bearings, to make it rigid. The top length of shaft casing is provided with a filtering means, which allows part of the water pumped to enter the shaft protecting casing after it is filtered. Passages are provided through each bearing to allow the lubricating water to circulate downward and out into the well through ports in the top of the lower pump. This method of lubrication prevents contaminating the drinking water with oil or grease.

JUST FOR FUN

DID YOU GET IT? YES? NO?

Editorials

Unscientific Design

A fault common with many designers is failure to understand and appreciate shop and manufacturing conditions that modify the cost of construction. Most drawing offices now have adopted standards for certain parts of their machines, but there is still much improvement that might be effected. The designer should have had considerable shop experience to understand all conditions, but this is not always possible.

Examples of unscientific design can be readily called to mind by anyone who has handled machinery. Frequently the design is such that very intricate patterns, cores and molds are necessary to make a casting, when a full knowledge of foundry practice would have enabled the designer to greatly simplify the pattern and the cores necessary. Furthermore, there are more bad castings when complicated patterns are used than with simple forms.

Designs of machine parts are probably influenced more by local shop conditions than by many other considerations. The designer alters his layout so that the tools on hand can be used and special tools avoided. Designers also frequently put angles in place of dimensions of certain parts of their drawings, when the workman has neither a suitable protractor to measure such angles nor often room to use one if he had it.

One often finds in assembling machines that no room has been left around bolts for wrenches and they must be tightened by chisels. This was frequently the case in the early designs of steam turbines. Sometimes bolts are placed in inaccessible places and consequently are never properly tightened. After the machine is finished, the operator sometimes finds parts that are difficult to lubricate properly and that are practically inaccessible for cleaning and inspection.

Mistakes in design are also frequent in power plants. A contractor for a Western plant recently closed a contract on a cost-plus-percentage basis. On looking over the designs for the condenser intake and discharge tunnels, which were large and long, he pointed out to the engineers that an alteration of the design to allow the repeated use of standard forms would cheapen the job by over a thousand dollars, as well as permit more rapid construction. The suggestion was immediately adopted, although the contractor would have made more money by following the original design.

An elaborate tunnel-heating system was installed by a large Eastern institution. The tunnels were large and well lighted and ventilated; but the designer crowded his heating pipes so close to the wall on one side that bolts in flanges were not accessible. It has since been a difficult task to repack the glands on the expansion joints or to replace gaskets in the flanged joints. These conditions were entirely unnecessary. The location of piping in inaccessible places is a common failing of designers.

Enough illustrations have been cited to emphasize the point that it is always advisable to give the most careful consideration to shop or plant conditions. This can generally be done by having the superintendent or chief engineer pass on the designs. Let him offer his ideas, examine drawings and make criticisms. Mistakes can often be avoided in this manner. The plan is also truly scientific, for the designer is collecting all available knowledge to use in his new plans. Consultation with the practical man also has a distinct psychological value, for if he has had a hand in the designs, he feels a certain sense of responsibility in their proper execution and will therefore maintain a friendly and helpful attitude toward the work when in progress of execution, instead of assuming the role of a critic.

Ash Handling in Power Plants

Efficiency in power-plant operation requires low labor cost per unit of output. Very close attention is often given to labor in handling coal, but in many plants the handling of ashes has not received the same careful attention. One frequently finds that the laborers, possibly because they are paid the lowest wages in the plant, are simply turned loose to dig out the ashes from awkwardly built ashpits and to truck them out of the way. Often a little thought on the part of the management would lead to the construction of more convenient and more accessible pits and the labor and cost of handling the ashes would thus be reduced.

How many engineers have ever stopped to show the ash handler the best method of hoeing out his ashes or the easiest way to shovel them onto a barrow or truck? How many barrows have been purchased that have been designed so that the laborer lifts as small a portion of the total load as possible and with ball or roller bearings to make them run easily? How many have provided special shovels and hoes for ash handling?

Some may ridicule the idea of giving much consideration to ash handling, for a laborer will be needed for this purpose and if he is paid for all day, it may not be thought worth while bothering about his work so long as it is finished during the allotted hours. But if he can do the same work in half a day, then he is available to help on other jobs, such as trucking coal, cleaning boilers, or simply keeping the boiler room clean and neat.

There is also the question of handling the ashes outside of the plant. Frequently they are allowed to accumulate until no more room is available. Then they are trucked away after being shoveled onto the trucks. This makes double handling of a dirty, dusty material. Has full consideration been given to possible methods of disposing of the ashes as quickly as formed? A little planning may make this possible even in small plants. Of course, where large quantities of ashes are handled, the designers of the plant usually give careful consideration to the disposal and one finds conveyors, industrial

railroads, vacuum systems, etc., installed and carefully watched, all tending to lower the cost of this work. However, it may be possible by the development of suitable ash disposal to convert a cost against power into a profit for the plant if local conditions are favorable.

Influence of Boiler Scale

Everyone concerned with the running of a boiler plant realizes more or less definitely that scale is an enemy to good service, both because of the physical injury from overheating that is liable to result and of the impaired efficiency of the heating surface. But being concealed from view in the ordinary course of operation, scale happens to be one of those clogs on economy whose effects appeal only in a remote way to many minds. The deposits may gather so insidiously, and the corresponding increase in the quantity of fuel consumed may proceed so gradually, that in many cases the true condition is not fully appreciated—the increased cost of operation observed being perhaps largely attributed to other causes than scale.

To what extent does boiler scale figure in the annual cost of maintaining a steam-power plant? Notwithstanding that percentage figures purporting to give definite information on this point have been tabulated, the question still seems to defy an answer that would be even approximately accurate in all cases. Hardly any two of the tables compiled by different authors agree upon the loss of fuel that will result from a given thickness of scale. One authority will charge a minimum loss of ten per cent. to scale of a certain thickness, while the conclusions of others will range from this figure up to twenty per cent. for the same quantity of scale, the diversity observed in the tabulated results suggesting their derivation from experiments with boilers of different types and under radically unlike conditions.

The influence of scale upon boiler economy is something that admits of no hard and fast rule. It depends altogether upon the conditions existing in each case and upon the physical properties of the scale. In charging a fuel loss of a certain amount to a certain thickness of scale, the presumption is that that particular thickness of scale is uniformly distributed over the entire area of heating surface; and this, as every practical engineer is aware, is a condition that rarely, if ever, exists. In many horizontal return-tubular boilers that receive scant attention in this regard, the furnace sheets may show a hard scale scarcely more than one thirty-second of an inch thick, the corresponding ends of the tubes being incrusted with a coating of relatively soft porous scale considerably thicker, while at the back end the scale may be so thick as to fill the space between the tubes. To figure on the comparatively thin scale observable through the front manhole in cases like this could obviously result in valueless conclusions.

Prevention of scale is one thing and cure is another. Whether the cost of prevention, coupled with the advantage of keeping the surfaces always clean, will show a balance over the cost of cure, when coupled with the disadvantage of permitting the surfaces to gather scale for a time, is the main question with many steam users. Engineers may still be found who contend that a little scale on the water surfaces of a boiler is beneficial; but the majority will agree that it is best to prevent the formation of scale altogether. Again, there are power-plant owners who seem to think it the more profitable plan to let the scale accumulate for a while and then remove it by mechanical means. With these the matter simmers down to a comparison of the cost of continued prevention with the cost of periodic removal plus the attendant loss of fuel and the danger of possible damage to the boiler.

There are so many varieties of scale that a method of treatment that will prove satisfactory in one case might fail utterly in another. Kerosene used to be one of the main resources of operating engineers in ridding their boilers of scale. But when applied effectively, the penetrative quality of kerosene generally brought disagreeable consequences in addition to the good accomplished. It would get beneath the scale, but it would also force its way between jointed surfaces and thus cause troublesome leaks. By pouring a half-gallon or so of kerosene into a badly scaled boiler and then cracking the feed valve so that a day and a night would elapse while the boiler was filling, thus giving the oil ample chance to exert its penetrative power, the operator would generally have no difficulty getting the scale away from the sheets and tubes the next time he cleaned out. He would, however, be fortunate if he did not also find considerable repair needed in the way of rolling tube ends, calking seams and rivets and replacing leaky gaskets.

Other materials that have been regarded as universal panaceas for scale are gum catechu and various forms of soda. Indiscriminate use of these substances has often led to trouble. The chemistry of boiler water has been investigated to a point where it is not necessary for an engineer to resort blindly to the use of any of these in their raw state for getting the scale out of a boiler in which it has been allowed to accumulate and for preventing its further formation. But no matter what method of treatment is used, a sharp lookout must be maintained to guard against foaming and the consequent impairment of engine lubrication, resulting in damage to pistons and valves, as foaming may result from the presence of decomposed scale in the water.

Our readers will be gratified to learn that the Ohio Board of Boiler Rules, at a special meeting held December twenty-first, unanimously adopted the American Society of Mechanical Engineers' Boiler Code, in so far as it relates to the construction and installation of new boilers, these rules to take effect July 1, 1917. This action is in accordance with the resolutions adopted at the American Uniform Boiler Code Congress held at Washington, D. C., December fourth to sixth. With Ohio in the van and prospects of Massachusetts soon falling into line, the situation assumes an encouraging aspect for the new year.

The present outrageous prices of coal are offering exceptional opportunities to the promoters of coal-saving nostrums. Take it from us there is no substance which can be sprinkled upon or mixed with coal which will be any better than the same money's-worth of more coal.

The complete report of the Hall of Records Test has at last been printed. But then—all things come to him who waits.

Correspondence

Coal and Ash Handling at Grundy Plant

L. V. Curran in the Nov. 14 issue, page 695, calls attention to the fact that in my article about the coal- and ash-handling equipment at the William H. Grundy plant (page 480, Oct. 3, issue) I did not mention any power costs. I am glad that Mr. Curran wrote this letter, since it gives an excellent opportunity to show what a small item the power costs are in handling coal and ashes by machinery. As a matter of fact, in my article the item of 3c. per ton for unloading coal from cars is intended to include the cost of conveying the coal to the bin although I neglected to mention this fact.

Mr. Curran has made a little slip in working out the kilowatt-hours from the formula Kw = pull in lb. × speed in feet per minute × 0.746 ÷ 33,000. He takes this, for the lifting of the coal only, as (200,000 × 72 × 0.746) ÷ 33,000 = 325.52, the 200,000 being the number of pounds of coal lifted in 24 hr. and the 72 being the number of feet of lift. What he really obtained is the number of kilowatts required to lift the total daily amount of coal in one minute. If, instead of lifting this load in one minute, it was lifted in an hour, it would take one-sixtieth of this amount of power, or 5.4 kw.-hr. per day for the actual lifting of the coal. As the figures for the amount of power required for elevating the ashes and the total number of kilowatt-hours per day are obtained similarly, these should also be divided by 60.

The usual way of figuring the kiolwatt-hours per day for a conveyor is to get the horsepower necessary to operate the conveyor, obtained from the pull necessary to move it and the speed, reduce this to kilowatts by multiplying by 0.746, and multiply by the number of hours that the conveyor would be in operation each day. In this case the capacity of the machine is 25 tons per hr., or 833 lb. per min. and the speed is 30 ft. per min. Dividing this 833 lb. lifted per minute by the speed of 30 ft. per min. gives the required amount of coal per foot as 27.7 lb. Multiplying this figure by 72-ft. lift gives 1,994 lb. of coal being lifted at any moment at a speed of 30 ft. per min. This gives 1,997 × 30 × 0.746 ÷ 33,000 = 1.35 kw. for just the lifting of the coal. In order to handle 100 tons of coal the conveyor would have to operate for four hours. So that if we multiply the 1.35 by 4 it gives 5.4 kw.-hr. per day, the same as dividing Mr. Curran's result by 60.

As a matter of fact the motors used are one of 15 hp. for operating the crusher, feeder and apron conveyor, and one of 7½ hp. for operating the Peck carrier. After the machinery is started, this much horsepower would not be required, but larger motors are necessary in order to get the starting torque for overcoming the static inertia. If we figure 20 hp. for 4 hr. per day for handling coal and 5 hp. for 4 hr. per day for handling ashes, it gives a result of 100 hp.-hr. per day, or 74.6 kw.hr. per day for handling both coal and ashes. Taking Mr. Curran's figure of ½c. per kilowatt-hour for power gives a daily power cost of 37.3c. and a yearly power cost of $136.14. This does not take into account the power required for moving the traveling weighing hopper, but this is so small as to be practically negligible. The cost of 3c. per ton for unloading coal from cars and handling to bins would therefore seem to be ample. With the trestle equipment the power cost would probably be offset by some additional labor required for trimming the coal out from underneath the trestle at times when the trestle becomes filled up so that it is necessary to push the coal out sideways in order to be able to dump the cars.

Philadelphia, Penn. HENRY J. EDSALL.

Oil-Burning Boiler Furnace

In reply to A. C. McHugh's criticism in the issue of Oct. 31, page 630, of the oil-burning furnace described by me Sept. 26, page 464, I would say that results are what count and since taking charge of this plant, in spite of increased load, I have cut down the fuel-oil consumption 40 per cent. This saving is due in large part to furnace and burner improvement. In the last five years I have operated three oil-burning plants. In one the furnace was arranged exactly like that recommended by Mr. McHugh, which I suppose is the first type of oil-burning furnace designed. In fact the only departure from the coal burner is the bricks on the grate. I have never seen anyone who could give a good reason for a grate in an oil-burning furnace. From experience with both I am convinced that the furnace described by me is an improvement. As to blistering the boiler, I do not think there is any more danger with one than with the other.

We have never had anything like a blister, and during peak load, which lasts about six hours a day, the boilers are crowded. I have seen the brick arches at almost the fusing point, yet very little flame would go over the top of the arch where it would come in contact with the boiler shell. The heat is intense certainly, but that is the case under any boiler that is being forced. It is only necessary that the boiler be kept clean, which should be done in any case. If this is done, there will be no need to worry about blisters or bags.

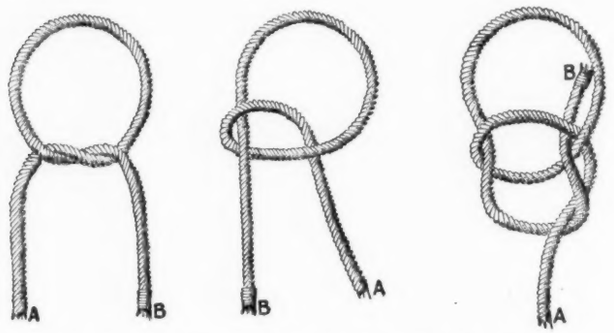
Another advantage of this furnace is that all danger from furnace-gas explosions, caused by the fire going out, are eliminated, as the highly heated arch burns all gases as fast as generated and the fire may be started with perfect safety, after it has been out an hour or more, by merely opening the damper and turning on the steam and oil. It will ignite instantly.

As to the burner, I have used a number of so-called standard makes and have found only one as satisfactory as the home-made ones I am using now; still I am not altogether satisfied with this burner and stand ready to adopt a better one when found. We should always look and work for something better in everything that relates to power-plant efficiency. W. G. CAMP.

Ash Fork, Ariz.

Tying a Bowline Knot

On page 530 of the issue of Oct. 10, Mr. Clark shows how one may go wrong in tying a bowline knot, by putting the end of rope through the loop the wrong way. Out here in the range country we have a way of avoiding any mistake in that part of the process. Here it is:



THREE STAGES IN TYING A BOWLINE KNOT

Pass the end of rope around the object you intend to tie to. Pass the end around the rope the same as if you were tying the first half of a square knot, as shown in the illustration. Leave the rope *A* slack and give the end *B* a quick jerk, at the same time passing *B* over *A*. This will cause the rope to form the loop with the end of rope through it. Then pass the end around under the rope and back through the loop and draw upon both, completing the knot. This method is quicker than the other, and the mistake pointed out is impossible.

Gordon, Neb. W. F. HANKINS.

A Heating-System Anaylsis

In the article copied from the bulletin of the National District Heating Association entitled "A Heating-System Analysis," on page 800 of the issue of Dec. 5, some of the statements are unfair to the vacuum-return system and are not borne out by modern practice.

One is that considerable water is necessary to condense the steam that leaks past the radiator traps. There are on the market several makes of traps, through which the leakage is so small that no injection water is required.

No mention is made of the possibility of making a considerable saving in mild weather by carrying a low pressure on a vacuum system or by the use of modulating valves. With a gravity system it is almost impossible to vary the temperature of the radiators to an appreciable degree, so that in mild weather many windows in the building are opened, with a consequent waste of steam. There are few gravity systems that are not noisy at times, especially when turning on the steam in the mornings, while a well-designed vacuum system is practically noiseless. The vacuum system does away with troublesome air valves, which frequently leak steam and water and do damage.

It is claimed that 5 per cent. of the steam supplied to the system as a whole is wasted by the vacuum pump, but as the vacuum pump is always exhausted into the system, the steam chargeable to it is practically nothing. A direct-acting steam pump converts into work only about 2 per cent. of the heat supplied to it, so that it is nothing more than a reducing valve for supplying low-pressure steam to the system.

An "economy" coil is shown on the gravity system for recovering some of the heat from the condensation. Why was not an economy coil shown also on the vacuum system? Such coils are sometimes used on systems where there is trap leakage, to do away with the necessity of using injection water. DALLAS F. GRAHAME.

Toronto, Ont., Canada.

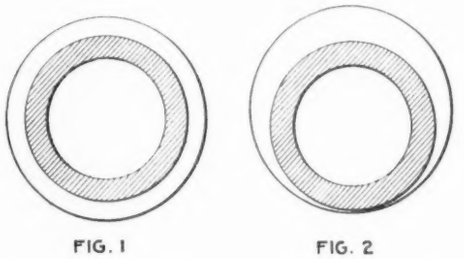
[The article referred to was intended to apply particularly to district or central heating.—Editor.]

Economy in the Use of Packing

The cost of packing constitutes an item of no small amount in power plants, and economy in its use is well worth the attention of every engineer. The superintendent of a plant told a packing salesman recently that he would rather have the money for packing that had been wasted in that plant than that spent for packing actually used. The salesman admitted he was about right.

Gaskets for flanged fittings are often cut by laying a sheet of packing on the flange, taking pains to have the sheet extend over a little on all sides then hammering the packing over the edges of the flange, thus cutting a ring the full width of the flange besides wasting a little packing all around it. As has often been pointed out, a ring that will fit just inside of the bolt holes really makes a better gasket than one that covers the whole flange, and it costs less than half as much. One man cut a cylinder-head gasket that cost thirty cents. Another used up material that cost $1.30, and the job was no better.

Another source of waste is in improperly cutting gaskets from left-over circular pieces. Many engineers will take a circle larger than needed, cut the center out and leave the remainder on the fitting or whittle it off in such a way as to render it useless. The proper way is to cut out of the circle a concentric ring just the size needed,



REDUCING WASTE IN CUTTING GASKETS
Fig. 1 the right and Fig. 2 the wrong way

leaving a ring outside and a circle inside for use on fittings of other sizes. Figs. 1 and 2 show the right and the wrong way; enough packing is wasted in the latter case to make a gasket of a larger size.

Tubular packing is a very economical form to use for flanges, and there need be no waste, as short pieces may be spliced together to make any size of gasket desired. By running a copper wire of about No. 16 gage inside of the tubular gaskets, they are suitable for pressures up to 125 lb. The wire should go all the way around and lap a little on the side opposite the splice in the gasket. The ¼-in. size is large enough for flanges up to 5 in. and ⅜ for larger sizes. The ⅜ size weighs over twice as much as the ¼-in., so the smaller should be used wher-

ever suitable. Engineers need to learn that there is such a thing as good enough and that if a thing is good enough there is no use in trying to make it better, because it cannot be made better. All over enough is usually wasted.

In cases where it is difficult to keep the tubular gaskets in place while putting the work together, they may be tied in position with a soft string or small copper wire threaded through the bolt holes.

An engineer, in putting up an 8-in. steam line, was skeptical about tubular gaskets and used several other kinds in preference. However, he ran out of other material and was forced to use one tubular gasket. When the pipe line was put in service, the tubular gasket was the first to be made steam-tight and it remained in use after all the others had been renewed.

If engineers will figure out the cost of the packing they use and think what they would do if they had to pay for it themselves, their packing expense will be cut in half. G. E. Miles.

Green River, Wyo.

Desires Refrigeration Regulations

Your editorial suggestion, in the Nov. 28 issue regarding regulation of installations of refrigeration plants is pertinent and opportune. It would seem the better part of wisdom for the interested parties to run to cover so conveniently near, for no doubt, unless they themselves point the way to greater security, it will be taken up by interests less competent but more powerful, resulting in a confusion of local regulations difficult to efface, once established.

The art of manufacturing and testing ammonia containers of a dependable character should entail no hardship. Hydrostatic testing to the required margin of safety is simple and the requirements should be rigid. The certification of such test before witnesses is of the same nature, as is in use along other lines and has stood the test of long and legal usage.

It is impossible to get married without a license, or to procure the arrest of a person without a warrant, yet anyone can assemble the lowest class of junk into a refrigerating plant and install it in the midst of a busy hive of human beings with the inevitable results recently chronicled. It is folly to think of allowing a continuation of such conditions, therefore it would look better to find the interests signing the pledge when the police arrive.

Passaic, N. J. E. E. Clock.

Peep-Holes in Furnace Doors

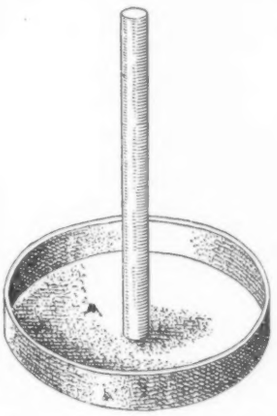
The losses caused by excess air in the operation of boiler furnaces is too well known and understood to call for comment, but to avoid excess air is at times difficult. I was surprised to learn how often the (hand-fired) furnace doors were opened to see the condition of the fire, when no coal was added. In an equipment with eight doors where four furnaces were in use, the doors were opened an average of ten times an hour, or about eighty times in an eight-hour shift. The doors were removed, and a 1½-in. hole was drilled through the upper half of each and a slotted hole cut out of the door lining. The hole in the door was covered with mica, which was held in place by a large washer turned out on one side to receive it and secured to the door with two small capscrews. This made a handy peep-hole through which to watch the fire without opening the door and admitting a large volume of cold air into the furnace, as under the former condition.

The firemen were then instructed to watch their fires and add fuel only when the damper was closed. This resulted in raising the percentage of CO_2 about 1½ per cent., with a corresponding reduction in coal used, or large returns at small cost. Henry W. Geare.

New York City.

Substitute for Regular Valves

It is sometimes difficult to get repair parts when located in a country power plant. The way one man overcame such a difficulty in getting new pump valves, when the old ones had been ground down at various times until they were so thin that they warped and the stem holes were enlarged so that considerable water was lost through, them, was as follows: A cardboard box lid, the same diameter as the valves, was secured, and an iron rod the diameter of the valve stem was placed in the center, as shown in the illustration. A mixture of lead and babbitt was melted and poured into the lid. This process was repeated for the required number of valves, several cardboard lids being used in the operation. A sheet of fine sandpaper was tacked down on a board, and the castings were rubbed over it until a fairly smooth face was secured. These valves gave such good service that they were left in permanently. When they started to leak, they were simply removed and "sandpaper faced" again.



MOLD FOR PUMP VALVES

Kingston, Ont., Canada. James E. Noble.

Flexible Coupling Trouble

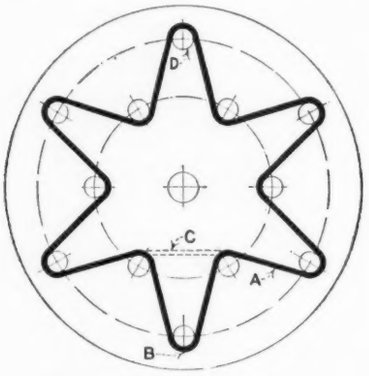
One day I got a "trouble call" to see a 35-hp. induction motor connected by flexible coupling to the countershaft of a large wood planer.

"The motor is out of balance and jumping as if it was going to pull the floor up, so I shut 'er down," I was told. I looked it over, but could see nothing wrong. I turned the motor and countershaft over by hand, and all turned free and true. I then turned on the "juice," and "out-of-balance" was no name for it. The countershaft mounted in the frame of the massive planer ran perfectly quiet, but the motor, set on the plank floor, was springing the flooring fully a quarter of an inch. I noticed, though, that when the current was shut off, the vibration ceased even though the motor was still at nearly full speed; but when I turned the current on again, the vibration was as bad as ever. This set me to investigating the flexible coupling, and there I found the trouble.

In the illustration the pins on the inner circle belong to the motor half of the coupling and those on the outer

circle to the countershaft half. *A* is a leather strap used to make the flexible connection. This strap had broken, and being then too short, the millwright had left it off the pin *B*, so it would reach and laced it on as shown by the broken line *C*. This practically cut out three pins and gave a drag crank effect to the coupling, which caused the trouble while under power.

This effect is easily understood by imagining a man in the place of the motor turning the shaft by means of a



TYPE OF COUPLING USED

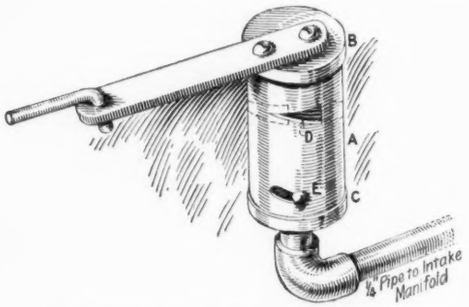
crank—his feet made fast to the floor. On the up half of the crank revolution the man would be pushing down with his feet as much as he was lifting with his hands, while on the down half he would be lifting as much from his feet as he was bearing down with his hands.

I slipped the strap off pin *D*, and the whole trouble was cured. The motor was in no way "out-of-balance," but the power transmission was. R. Mamly Orr.

Vancouver, B. C., Canada.

Auxiliary Air Attachment for Gasoline Engine

The illustration shows an attachment for letting in air above the carburetor of gasoline engines, similar in principle to the numerous types sold for automobile use,



REGULATOR FOR AIR INLET

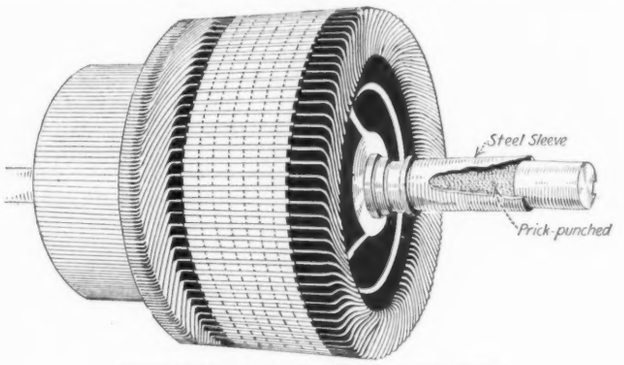
but without the nickel plating and high price. It may be used for either stationary or marine engines and should result in a saving of fuel, for the admission of air above would decrease to some extent the "suction" through the carburetor. The apparatus described is used on a two-cylinder engine running on distillate. A quarter-inch pipe is tapped into the intake manifold, and when the engine has run long enough to warm up, the auxiliary valve is opened wide, giving an opening into the manifold the full size of the pipe. This does not affect the speed of the engine, but results in an appreciable saving of fuel. It is thought that a still larger pipe could be used, although this has not yet been tried.

The valve consists of two short lengths of brass tubing, one sliding easily within the other, the outer piece in this case being about an inch in diameter. This piece *A* has a brass cap *B* soldered to the upper end to which a control handle *F* is fastened. The inner tube has a cap *C*, which is drilled and tapped for a quarter-inch iron pipe. After the two tubes are fitted together, a small screw is put in at *E* and a slot cut in the outside tube, the screw and slot keeping the tube from slipping off and from being turned too far. Three triangular slots *D* are cut through both tubes with a drill and a three-cornered file to control the quantity of air admitted. When the outer tube is turned in one direction, the slots are wide open, but when turned in the other direction, they are completely closed. The valve is shown in an upright position so that kerosene for cleaning or gasoline for priming may be squirted in from an oil feeder.

Oakland, Calif. H. H. Parker.

Repairing an Old Motor Shaft

It being impossible to buy a new 10-hp. 110-volt direct-current motor and obtain an immediate delivery, a second-hand motor of the same size was purchased. This old machine was in pretty bad condition; both bearings and the shaft at the pulley end were badly worn. If the



HOW WORN SHAFT WAS REPAIRED

part of the shaft in the bearing had been made larger than in the pulley, it could have been turned to a smaller diameter and the bearing bushed and reamed to fit it. However, the bore of the bearing and pulley were of the same size; therefore, if the shaft was turned to a new surface, it would have to be turned the entire length, as the bearing was of the one-piece type. This would require the pulley to be bushed, weakening the shaft, which was not any too strong to start with. The method of overcoming this condition was as follows: A piece of machinery steel ⅜ in. larger in diameter and ½ in. longer than the bearing was reamed to a slightly smaller diameter than the shaft. The worn part of the shaft where this sleeve was to fit was prick-punched around the small part, as shown in the figure, and the sleeve was heated and forced into place by a hand press. The sleeve was then turned down until it was ⅛ in. larger than the original shaft. This permitted the old bearing to be used by reboring and reaming. The bearing for the commutator end was bored out and a bronze bushing forced in. To make sure that it would not work loose, the bushing was soldered in place with an all-tin solder, then bored and reamed.

Washington, D. C. R. L. Hervey.

Inquiries of General Interest

Pump Discharge to Open Tank—Will more power be required to pump water through the bottom of an open house tank when the tank is full than if the water is carried by a pipe over the top of the tank? W. H. R.

Less power will be required to pump the water through the bottom of the tank, for by extending the pipe to the top of the tank the pump would be required to work against greater head pressure required for overcoming friction of a greater length of pipe, and unless the end of the pipe were dropped so as to discharge below the surface of the water in the tank for obtaining a siphonage effect, the pump would be required to work against a greater static pressure.

Transmission of Heat Through Bare Steam Pipes—What is the rate of conduction of heat from uncovered steam pipes to the atmosphere? R. G. T.

The coefficient of heat transmission through bare steam pipes varies with the pressure of the contained steam and also with the activity of circulation of the air surrounding the pipe. Assuming the air to be still, the conductance per square foot of external pipe surface per hour per degree F. difference of temperature of the steam and of the surrounding air is about 2.1 B.t.u. for steam at 5 to 6 lb. gage and about 3 B.t.u. for a pressure of 90 to 100 lb. gage. It is common practice to allow 2.5 B.t.u. per square foot per hour per degree difference of temperature for all pressures.

Relation of Valve Travel to Lap and Port Opening—Is the travel of a valve equal to the width of the port plus the lap of the valve? J. K.

The travel of a slide valve is the distance through which the valve moves, first in one direction and then in the other. Before the steam port can open, the valve must travel a distance equal to the amount of the lap, and when it is completely uncovered, the width of the port plus the lap will be equal to the travel of the valve, provided the movement is only just far enough to uncover the whole of the port. But if the port is so large that a part of it is not completely uncovered, the travel will then be less than the width of the port plus the lap of the valve; or if the valve moves farther than necessary to completely uncover the port, the travel of the valve will be greater than the width of the port plus the lap of the valve.

Induction Motor Slip—What is the meaning of the term slip when applied to the characteristics of an induction motor, and how is it obtained? W. R. B.

The slip of an induction motor is the difference between the speed of the rotating magnetic field of the stator and the speed of the rotor. It is usually expressed as a percentage and is found by the formula,

$$\text{Per cent. slip} = \frac{(S - R) \times 100}{S}$$

where S = the speed of the stator magnetic field and R = the speed of the rotor. R is obtained with a speed indicator and S is calculated by the formula,

$$S = \frac{120\ f}{P}$$

where f = the frequency of the circuit supplying the motor and P = the number of poles in the stator. The slip of well-designed induction motors varies from about 2 per cent. on large machines to 8 per cent. on small machines.

Meaning of Superheated Ammonia—What is meant by superheated ammonia gas, and what is the source of the superheat? H. B. G.

A gas or vapor at any pressure is said to be saturated when its temperature and pressure are the same as the evaporation temperature corresponding to its pressure; or as commonly expressed, when it is at the temperature of the boiling point of its liquid under the same pressure. When for any stated pressure the temperature of the gas or vapor is higher than the boiling point of the liquid for the same pressure, the condition is then said to be superheated. The number of degrees by which the actual temperature exceeds that of the boiling point corresponding to the pressure is called the "degrees of superheat," and this term is used to designate the amount of superheat.

In an expansion system of refrigeration the ammonia gas leaves the coils slightly superheated, and when the gas enters the compressor, it is superheated by the warm cylinder walls; that is, it attains a temperature higher than the boiling point of ammonia that corresponds to the same pressure.

Use of Different Steam Pressures for Jacketed Kettle—Can water be brought to a boiling point or made to boil more rapidly in an open steam-jacketed kettle supplied with steam at 80 lb. gage pressure than at 40 lb. gage pressure? J. D. W.

The greater the temperature difference between the steam supplied to the jacket of the kettle and the water to be boiled, the more rapid will be the transfer of heat. Steam at 80 lb. boiler pressure, or 95 lb. per sq.in. absolute, has a temperature of about 324 deg. F., and at 40 lb. boiler pressure, or 55 lb. per sq.in. absolute, the temperature is about 287 deg. F. Hence, with steam supplied at 80 lb. gage the water will be boiled more rapidly than with the steam at 40 lb. gage. It is to be understood, however, that in each case the pressures stated are actually maintained within the jacket of the kettle and that the condensation is drained off as fast as it is formed.

Working Pressure for Stayed Flat Sheet of Boiler—What would be the maximum allowable working pressure for the flat side of a locomotive firebox where the thickness of plate is ½ in. and the stay-bolts are pitched 6 in. by 7 in., with the stays screwed through the plates and riveted over? H. E. W.

The formulas of the A. S. M. E. Boiler Code for maximum allowable working pressure for various thicknesses of braced and stayed flat plates that require staying as flat surfaces with braces or stay-bolts of uniform diameter, symmetrically spaced, is

$$P = C \times \frac{t^2}{p^2}$$

where P = maximum allowable working pressure in pounds per square inch, t = thickness of plate in sixteenths of an inch, p = maximum pitch measured in inches between straight lines passing through the centers of the stay-bolts in the different rows, C = a constant, to be taken as 120 for stays, screwed through plates over ⅞ in. thick, with ends riveted over. By substituting C = 120, t = 8 and p = 7, the formula becomes

$$\text{Maximum allowable working pressure } P = 120 \times \frac{8 \times 8}{7 \times 7},$$

or 156.7 lb. per sq.in.

Determination of Pounds of Steam per I. Hp.-Hr.—How is determination made of an engine's economy in pounds of steam per indicated horsepower-hour? J. A. W.

When an engine must be operated noncondensing or with a jet condenser, then, assuming that all the steam from the boiler or boilers used goes to the engine tested, the weight of steam used can be determined from the weight of water supplied as feed water, the same as in an evaporative test of a boiler. This method of testing is known as a "feed-water test." When, however, this method is used, it is necessary to determine by a separate test the leakage of the boiler and the leakage and losses of steam by leakage and condensation of the piping from the boiler to the throttle valve of the engine. When determining such losses, the same boiler pressure must be maintained as when the engine is supplied for the test. For accurate engine tests it is preferable to determine the weight of steam used by directly weighing or measuring the exhaust as condensed in a surface condenser. In testing a condensing engine, the condensate may be weighed after being removed from the condenser by the air pump, or by gravity if the pressure in the condenser is atmospheric, as it would be if the engine is operating noncondensing. It is essential, however, that the surface condenser be tested for leakage before and after the test, and the leakage should be determined with the same vacuum in the condenser as when it is used in the engine test. The test should be commenced with a running start and continued with the same load for a period of at least two hours, and accurate indicator diagrams should be taken at equal intervals of 10 to 15 min. The number of pounds of steam used per hour divided by the average number of indicated horsepower will be the pounds of steam per i.hp.-hr.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Graphic Analysis of Steam Plant Design and Operation*

By R. J. S. Pigott†

SYNOPSIS—Describes the uses of the input-output curves of the main and auxiliary units and how these curves may be combined to assist in an analysis of design and operating costs.

In the design of new plants the graphic analysis is based upon the guaranteed water rates of the various pieces of apparatus proposed and will show the effect upon the cost of the plant of varying the combinations, such as substituting electric-driven for steam-driven auxiliaries. By means of the graphic representation of these facts in composite curves, many problems otherwise settled by professional judgment can be solved exactly in figures. In the operation of the existing plant, graphic methods are the most flexible and exact for analyzing the actual running conditions.

Individual Water Rates—The Willans input-output lines have been employed considerably in connection with the performance of steam turbines and steam engines, but for a complete analysis it is necessary also to have input-output lines for the boilers and all auxiliaries as well. Whether steam- or electric-driven apparatus is used, these lines are equally important. The input-output lines for the following

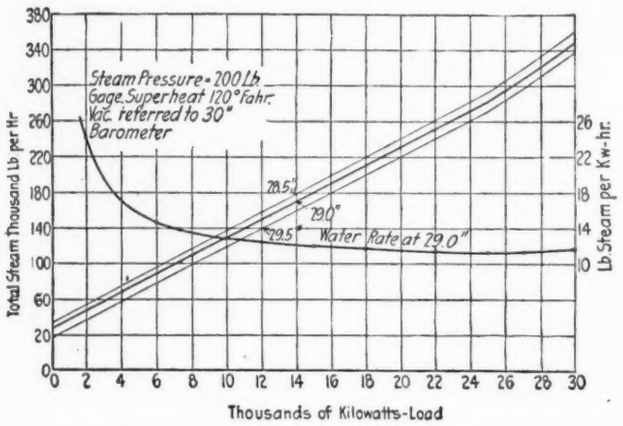

Steam Pressure - 200 Lb Gage. Superheat 120° Fahr. Vac. referred to 30" Barometer
28.5"
29.0"
29.5"
Water Rate at 29.0"
Total Steam Thousand Lb per Hr
Lb Steam per Kw-hr.
Thousands of Kilowatts-Load

FIG. 1. WILLANS LINES AND WATER-RATE CURVE 30,000-KW. INTERBOROUGH TURBINES

should be considered in a complete analysis: Main units, exciters, boiler-feed pumps, forced-draft fans, stoker drive and boilers.

Main Units—The Willans lines and water-rate curve for the 30,000-kw. Interborough Rapid Transit Co.'s turbines as guaranteed by the builder are shown in Fig. 1. Here it will be seen that the effect of variation in vacuum is a practically constant variation in the amount of total steam required over the entire range of operation. The percentage increase in economy, therefore, is greater at light than at heavy loads, a fact which should be considered in correcting input-output lines of the plant.

In the paper the author shows input-output lines for the various auxiliaries as mentioned; those for the main units and for one boiler only are given here in Figs. 1 and 2.

Boilers—In Fig. 2 the input-output curve of a single boiler is plotted from data obtained from actual tests. Heat units supplied are plotted against boiler output in myriawatts.[1] The efficiency at different rates of ouptut is also shown, the maximum being 78.5 per cent.; higher efficiencies are frequently obtained in single tests, but the values given represent the average of the best results obtainable in practice.

Fig. 3 shows the heat input per hour plotted against myriawatt output of 30 active boilers, 520 myriawatts rated capacity each. By plotting Fig. 3 to a large scale, it is found that the theoretically most economical method is to reduce to zero load with all boilers active, banking none. It is not feasible to operate in this manner, however, because of the difficulties of operating the underfeed stoker at loads less than 80 per cent. of capacity and the question of labor

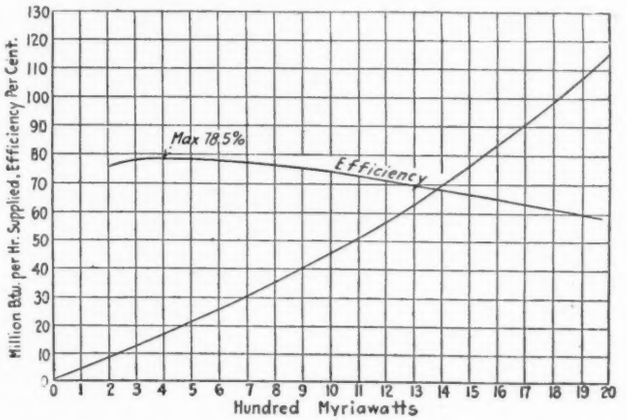


FIG. 2. OUTPUT-INPUT CURVE OF BOILER

costs. Also, opportunity is needed for making minor repairs without cutting a boiler out. Fig. 3 is therefore drawn on a basis of cutting out and banking as the load drops from 12,000 myriawatts at a rate of one boiler for every 400-myriawatt drop. This method is only slightly less efficient than continuous operation of all boilers.

Combined boiler-room auxiliaries, input-output lines, are given in the paper and show that for normal loads the steam

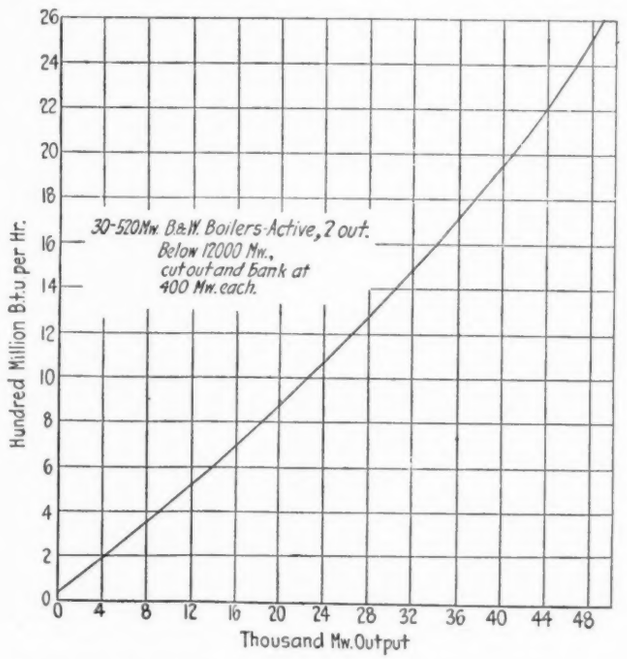

30-520 Mw B&W Boilers Active, 2 out. Below 12000 Mw., cut out and bank at 400 Mw. each.
Hundred Million B.t.u per Hr.
Thousand Mw. Output

FIG. 3. HEAT INPUT PLOTTED AGAINST BOILER OUTPUT

consumption of these auxiliaries varies little from 4.6 per cent. of the total steam generated.

The exhaust heat contents can be figured quite exactly because all except about 3 per cent. of the losses in a steam turbine reappear as heat in the exhaust. Therefore, heat in exhaust = total heat in steam supplied — (3 to 4.5 per cent.)

*Abstract of a paper before the December, 1916, meeting of the American Society of Mechanical Engineers, New York City.

†Mechanical engineer, Remington Arms and Ammunition Co., Bridgeport, Conn.

[1]The unit "myriawatt" has been suggested by H. G. Stott and approved by others as a unit of boiler capacity. It is 2 per cent. greater than the boiler horsepower and is equivalent to 34,150 B.t.u. per hr., the boiler horsepower being 33,479 B.t.u. per hr.

— brake horsepower × 2,547 — 3,200,000 B.t.u. The 3,200,000 B.t.u. covers radiation and leakage from the exhaust system. This radiation loss is figured, if possible, from area of pipe-line surface; but for quick figures, 2 per cent. of the heat of steam passing at full load will amply cover this loss, both on high-pressure and exhaust steam. Usually it does not actually exceed 1 to 1.5 per cent.

The total steam demand for the main turbine units in pounds per kilowatt-hour developed is shown in one of the curves of Fig. 4, marked "main units." All curves on this sheet are based upon 29 in. of vacuum, and the circulating water is assumed to be at a constant temperature from May 1 to Nov. 1. In Fig. 4 the curve marked "main units" is obtained directly from Fig. 1. To this is added 21,620 lb. of steam per hour per unit for the condenser auxiliaries. The resulting curve is marked "engine-room auxiliaries." Adding the boiler-room auxiliary steam demand, the third curve, marked "all auxiliaries," is obtained. There are also added 3 per cent. for starting, warming up and losses dependent upon load, and 25,000 lb. of steam per hour for pipe radiation, boiler dusting and leakage, which are independent of the load.

As pointed out by H. W. Flashman in 1911, it is noticeable that the economical point for cutting in units is not at the point of maximum steam economy of the unit under operation, but always at a higher point. This is most marked in

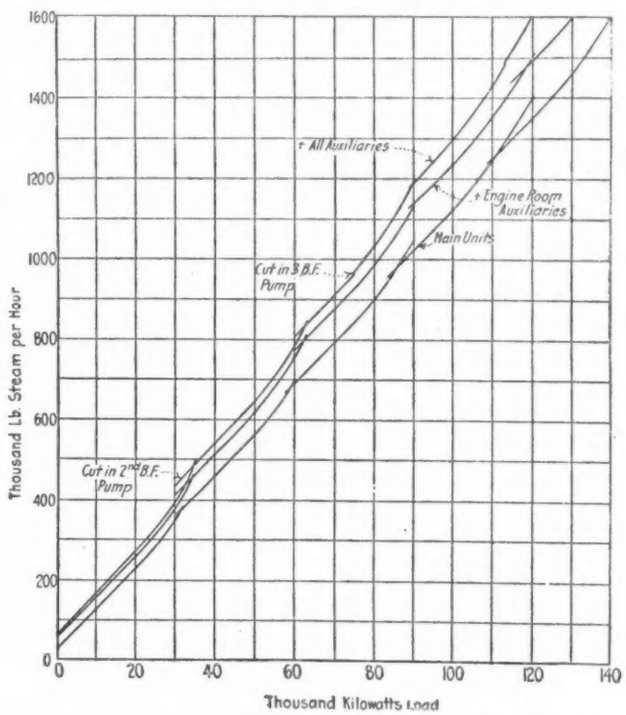


FIG. 4. COMPOSITE CURVES OF STEAM DEMANDS OF MAIN TURBINES AND AUXILIARIES

the curve showing the total steam consumption of the main units and all auxiliaries. Here it will be seen that the steam required starts on a more rapid rate of increase after 30,000 kw., yet if a new unit is cut in, approximately 8,000 lb. of steam per hour additional will be required at once. From the curves it is apparently inadvisable to cut in the new unit until the load has reached about 35,000 kw., when the total steam demand for the single unit has reached the total steam demand for the two units. Of course the deciding factor is the maximum capacity of the generator and the character of the load. But this feature proves that main units should seldom be designed for the best water rate at maximum load; that is, straight Willans line to full maximum rating. This is brought out plainly again in the Remington plant.

The author here takes up in an interesting way the effect of heat balancing between feed-water heater and the main unit, but unfortunately lack of space precludes its use here.

The choice between steam- and electric-driven auxiliaries can be definitely settled for any given set of conditions. The engine-room steam consumption will be that of the main units only, with the same allowances for losses varying with load and radiation, but the capacity of the main units is reduced by the amount used by the auxiliaries, the effect being to raise the input-output line. The greatest difference will appear in the boiler room, as there is no recovery of heat in the feed-water heater.

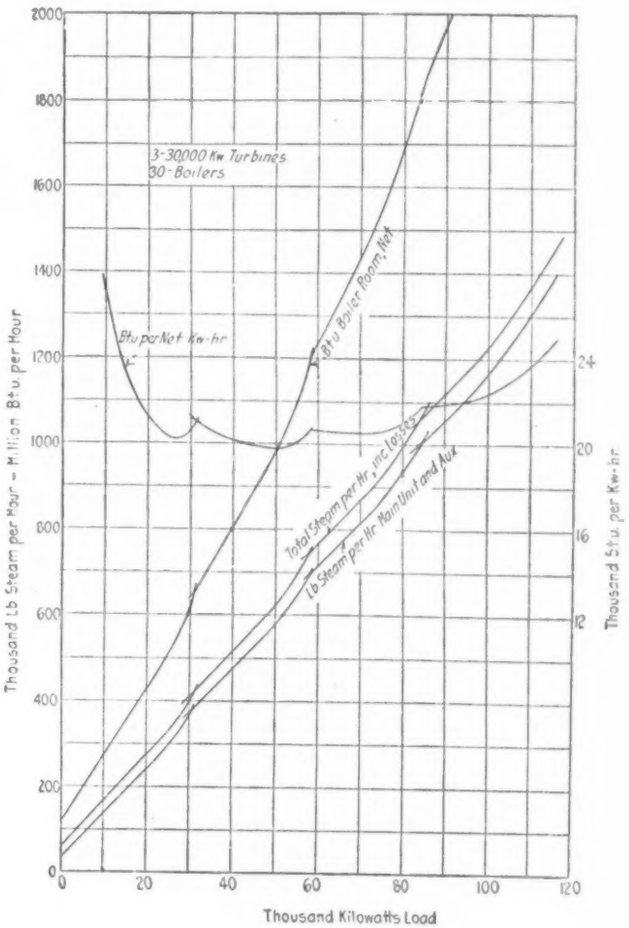


FIG. 5. SHOWS THE NET B.T.U. REQUIREMENTS PER HOUR

Fig. 5 shows conclusively that this arrangement is thermally less economical. In addition, the fixed charges are increased, the prices of all auxiliaries being higher with the electric drive, and the portion of the main-generator capacity and engine-room equipment used for auxiliary power supply also being charged. The saving in auxiliary piping is offset by the cost of wiring and switching equipment. The cost of the steam auxiliary station per effective installed kilowatt is $65, against $69.10 for the electric auxiliary station. As the simple electric auxiliary station is defeated on grounds of

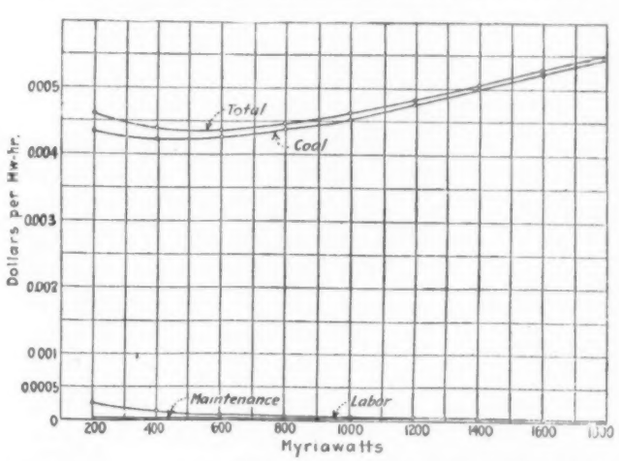


FIG. 6. COST PER MYRIAWATT HOUR TO OPERATE BOILER AND STOKER

both operating economy and investment charge, it can be eliminated from further discussion.

Let us now consider a separate turbine-driven generating equipment for serving motor-driven auxiliaries and exhausting its steam into the feed-water heater. An economizer will

also be considered. With feed water leaving the heater at 208 to 210 deg. F. and an average flue temperature of 450 deg. F. or less, the economizer is of doubtful value. Cold water cannot be used in the economizer because of the difficulty encountered with sweating, which causes clogging and renders the economizer inoperative. This is borne out by the experiences with the original Seventy-fourth Street Interborough Station in which all the auxiliaries except the exciters were electric driven and economizers were employed.

A short analysis will show the comparative value of this arrangement. Assume a load of 50,000 kw., which will give about the best economy for both steam and electric drive. The data are: Main steam units, 561,200 lb. of steam, auxiliary power requirements, 1,989 kw., 52,140 lb. of steam, including induced-draft fans and economizer scraper drive. In the steam-electric units, the temperature of the feed water would be raised from 70 to 155 deg. F. by the exhaust from the auxiliary turbines, and the possible rise in the economizer would be 147 deg. F. (limited by the fact that flue-gas temperature cannot be brought lower than 150 deg. F. above water temperature). The use of the economizer will lower the B.t.u. per kilowatt-hour to 18,000, as against 19,250 without it, or about 6.5 per cent. Economizers could be used with the all-steam units, but on account of the feed-water temperature, 208 deg. F., the saving would be only 4.1 per cent., making the B.t.u. per kilowatt-hour for this type of plant 18,540. The steam-electric plant, therefore, shows an improvement of 2.9 per cent. This, of course, is only at one load; the steam-electric combination does not necessarily operate more efficiently at all loads. The investment cost per kilowatt, however, is increased from $65 to $70.31. Whether it would pay to use the steam-electric combination, viewing it from the point of view of cost of power only, would depend upon the load factor. However, there are other items to be considered. There is an increase in the costs of labor and maintenance due to the extra turbines, generators and economizers. The electric-driven auxiliaries are no easier to handle than the turbine-driven auxiliaries, and flexibility is lost to a certain extent, especially in dry-vacuum pumps and circulating pumps Reliability certainly is not as good with the electric-driven units, as two additional links have been added to the auxiliary system. In the writer's opinion electric-driven auxiliaries should never be used for boiler feed and only with steam support for excitation. While dry-vacuum pumps and circulating pumps are less affected, from the operator's point of view, they are better off with turbine drive.

The best number of boilers and stokers to install for a given load curve can be determined readily. The cost per myriawatt-hour to operate a boiler and stoker is given in Fig. 6. The curves are based on the following data: Coal at $0.103 per million B.t.u. ($3.30 per ton, heat value 14,250 per lb. of dry coal); maintenance at $0.03 per ton of coal fired; labor, one operator at $0.40 per hour and one helper at $0.20

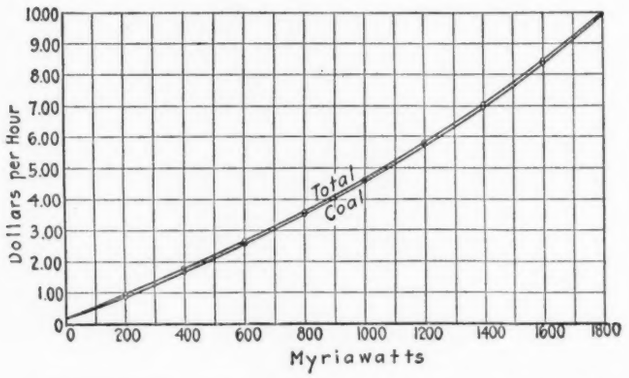


FIG. 7. TOTAL COAL COSTS PER MYRIAWATT HOUR

per hour for every 12 stokers. Fig. 7 shows the total operating cost per hour per myriawatt load and the total cost of coal. The total cost plainly follows the heat input-output line. In the paper curves are given which show the boiler-room cost lines in dollars per myriawatt-hour for different numbers of boilers in operation.

Using the total heat demand (of the plant), but separately for different numbers of boilers, shows that 30 boilers are more economical at light loads and 37 more economical at heavy loads, and for a typical day shows a net saving of $59.20 by the larger number of boilers. In a new plant, unquestionably, the greater number of boilers would not pay, as the fixed charges more than offset the operating saving. The actual case required stokers and equipment, but not boilers, as these were already installed. The comparison was that of 37 boilers in commission, 40 installed and 30 in service, 32 installed. The saving would not pay the fixed charges on th greater number of boilers even in this case. It is evident that the load factor is the controlling item: One must know the shape of the load curve as well as the peak loads and daily output. For a high load-factor plant the larger number of stokers, forced less, will generally pay.

The Remington plant[2] offered some interesting studies for graphical interpretation. Low-pressure steam at 15 lb. per sq.in. gage is used for industrial processes. Bleeder turbines are used, operating condensing in connection with a spray pond. The original plant had one 750- and two 2,000-kw. units, but was rapidly increased by the addition of two 4,000-kw. straight condensing turbines. The first two turbines are bleeders, and the third (2,000-kw.) is arranged for both bleeding and mixed-pressure service. This is very desirable, as about 125,000 lb. of steam per day is used for air compressing, the exhaust being available for industrial purposes or for the

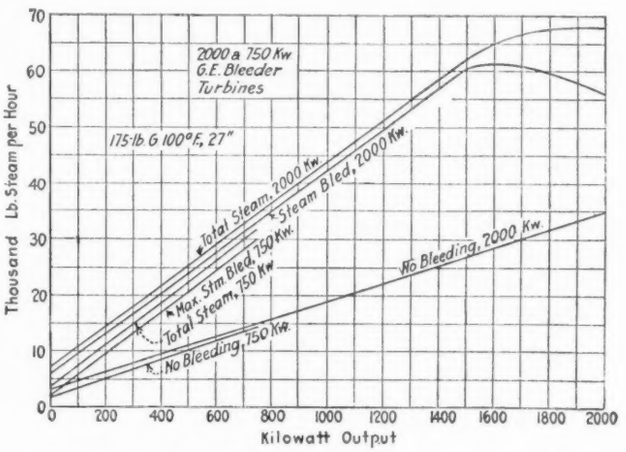


FIG. 8. SHOWING EFFECT OF BLEEDING STEAM FROM TURBINES

mixed-pressure turbine. Approximately 385,000 lb. per day of high-pressure steam for hammers, etc., and 90,000 lb. of low-pressure steam are delivered direct to the factory. In winter, all heating service is from the bleeder turbines, and the hot-water lavatory service is from them during the entire year.

The paper gives curves showing the Willans lines for the bleeder turbines, the effects of bleeding, etc., which are not given here. The effect of bleeding is very marked, especially at the lighter loads, when a large percentage of the total steam is bled. It is found that power can be made for 0.28c. per kw.-hr. in the winter months, increasing to 0.52c. in summer when the plant becomes practically straight condensing. This is with a ratio of average load to installed capacity of 19 per cent. and a ratio of average hour to maximum hour of 62 per cent. The figures actually obtained show 95 per cent. attainment of the foregoing.

Of course the plant is much underloaded at present, but the figures given will not be greatly altered by increase of load (shown by curves in the paper).

For rapid estimates it is evident that the curves presented in this paper will give some useful relations for analyzing operating records. For instance, the engine-room steam demand with all-steam auxiliaries balanced, might be represented with fair accuracy by the straight line $H = a + bL$, where a = the no load losses in the engine room (B.t.u. per hr.), b = the intrinsic B.t.u. rate of the engine room (B.t.u. per hr.), L = the kilowatt load on the plant, and H = steam demand (B.t.u. per hr.). This average curve smooths out the humps in the original line. It is found, by investigating eight cases, that the total heat input-output amounts between 0.30 and 0.80 load factor will lie on the curve of total B.t.u. plotted from the foregoing straight-line engine room demand and by use of Fig. 3 for any load curve. Variations of peak load within 15 or 20 per cent. are also found to agree. In short, the average heat consumption for any load curve can be taken from this chart as the amount corresponding to the average load without correction. This is of great convenience in predicting coal consumption for change of load factor.

Since the boiler input-output line is a second-degree curve, the variation in coal consumption with the load cannot be a straight-line function, although it may be nearly so for short ranges if the boilers are not used at high ratings.

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[2]Described in "Power," Mar. 7, 1916.

WALTER N. POLAKOV: The Willans-line method does not cover entirely other factors—namely, financial factors—that are as important and sometimes more so than the thermal efficiency of the equipment of the plant. In established plants of manufacturing concerns we see, as a rule, not only mismanagement, but also overequipment. In many cases the thermal efficiency may be very much sacrificed to gain the financial efficiency, and in some others it will be reversed; in other words, the picture will not be complete, even if the plant is not under operation, but only under contemplation for building, unless the various lines are supplemented by the method, also graphical if you like, of studying commercial and financial relations between the economy and efficiency of operation.

NORMAN G. REINICKER: As to the total water curve that Mr. Pigott uses, I have used that same scheme, without quite so much trouble, by plotting the feed-water demand as indicated by venturi-meter readings for the same load over various numbers of days, and I find that the effect of putting on another turbine will show up in that curve. It saves the testing of various machines and gives a rough calculation at least. The graphical method is valuable where you can predict the load curve accurately, but I have in mind the New York Edison Co. plants. It is very hard there to predict what we may expect. Therefore such a scheme is less useful to us. I am not sure I agree with Mr. Pigott regarding cast-iron economizers. The bad name they have received in the last three or four years is not due so much to the fact that they are cast iron, but because experience up to that time was always gained in installations where the feed pumps were reciprocating, and constant variations in feed-water pressure were quite common, and naturally the economizer received more shocks, and there was also the sudden variation in temperature which sometimes existed.

SHERWOOD F. JETER: Has anyone any record of economizers broken by pulsations from the use of a reciprocating pump?

NORMAN G. REINICKER: Positively, no; but apparently, yes. Some years ago few recording instruments were used, and we had no record of the feed-water pressure, as we relied on the word of the pump man. I have in mind one Edison plant, the Delray station, where the economizers did get a bad name, but I do not believe that it was because they were made of cast iron. They are behaving nicely with centrifugal pumps and with constant feed-water temperature.

JAMES W. PARKER: I have seen the pressures on economizers which were installed to carry 250 lb. run up to 400 lb. with centrifugal pumps, with no damage to the economizer. I would not like to see that happen with the old plunger type of pump. At all times it is desirable to generate as much auxiliary energy as possible in the heating of the feed water. If it is possible during the period of light load to take power for auxiliaries from the main bus and during the period of heavy plant load to transfer that from the auxiliary generator, if you have such an installation, to the main bus, one can use the main electrical system as a reservoir for the storage of energy, and at all times have the "cream" of the auxiliary steam turned into auxiliary energy.

ARTHUR M. GREENE, JR.: How can Mr. Pigott approach within 20 degrees of the temperature of the flue gas in the boiler and yet say that the flue gas at 400 deg. and feed water even at 210 deg. for economizer service is of little value?

R. J. S. PIGOTT: In this country practically every economizer is of cast iron. The difference in transmission between rough cast iron and that of steel boiler tube is pronounced, and one cannot work at the same rate of heat transfer at the same temperature difference. The third pass in the boiler is the most expensive.

Bureau of Mines Annual Report

In the latest annual report of the Bureau of Mines, just issued, Director Van H. Manning sets forth that the purpose of the bureau, in its fuel investigations, is to increase efficiency in the use of mineral fuels and to disseminate information about the nature and properties of the many available fuels, the facts about the process of combustion and heat generation and the best means to be employed for transmitting heat to the desired place, since current practice in the use of fuels is frequently based upon misconceptions as to relative fuel values, the nature of coal, the combustion process and the method of heat transmission. These studies find direct application and help to raise average efficiency in the general use of fuels.

Heretofore furnace design has been largely a matter of cut and try, resulting in many uneconomical misfits and in the production of objectionable smoke. During the last year work has been completed that makes possible the varying of furnace design in a rational manner according to the chemical properties of the fuel to be used and such studies as would help to remedy difficulties and make available quantitative values for the use of engineers in designing furnaces. A report of the results is ready for publication.

Combustion in the fuel bed of hand-fired furnaces burning coal at rates of 3 to 20 lb. per sq.ft. per hr. was also studied. This work was an extension of a similar previous study to the low rates of combustion, as in house-heating apparatus. Tests were completed with three kinds of fuel—Pittsburgh coal, anthracite and coke—and a report is in course of preparation.

A study of the combustion in a special furnace having a long combustion chamber, with the main object of determining the proper combustion space for burning any given coal without producing objectionable smoke, has been continued. These experiments are giving quantitative results of great value.

The object of the heat-transmission investigations is to determine the exact relation between the several factors governing the laws by which heat is transmitted from burning fuel to the water in a boiler. A knowledge of this relation will enable engineers to design boilers that will be more efficient as heat absorbers. During the year 1916 a series of 55 tests was made with special apparatus. The factors particularly studied were the influence of initial temperature of the furnace gases and their velocity through the boiler tubes.

Supplementary work done on commercial boilers consisted of a series of measurements of the temperature of gases as they passed through boilers of various designs.

Reports completed during the year and submitted for publication included those on combustion in the fuel bed of hand-fired furnaces, combustion of coal and furnace design, the effect of low-temperature oxidation on the hydrogen of coal and the change in weight on drying, moisture and spontaneous combustion in stored coal, the diffusion of oxygen through broken coal, and the absorption of methane and other gases by coal.

The investigations of both peat and lignite have been carried on and are of especial importance to those who live remote from coal fields but near deposits of these fuels, since little was previously known of the extent of the peat deposits or of methods of extracting, drying and preparing peat fuel at a profit and on a commercial scale. It has been shown that large areas of peat of value lie near iron mines in northern Minnesota now being worked entirely by coal shipped a thousand miles or more. Many of the peat deposits of the United States contain an unusually high percentage of nitrogen, have good fuel value and can be utilized by processes known to be thoroughly practicable on a large commercial scale.

The Bureau of Mines has, since its inception, had charge of the analyzing and testing of fuels purchased by the Government, amounting to between seven and eight million dollars. The bureau has been able to attack the fuel problems of the Government in only a small way, because of its restricted authority and the limited funds available, but has conducted investigations at some of the Government plants on request of the departments concerned and effected savings in several cases.

These results, which were attained in connection with only a small part of the fuel used by the Government, indicate the economies that might be realized by the purchase and utilization of fuel for the entire Government service under expert supervision. On a basis of $7,500,000 per year, a saving of 5 per cent., which is a conservative estimate, would amount to $375,000 a year.

In order for the bureau to coöperate most effectively with the various departments and bureaus of the Government in matters relating to the selection and use of fuels at Government power and heating plants, it will be necessary for engineers of the bureau to visit the plants in order to determine the best fuel to be used and what changes in equipment or method of operation are advisable and to obtain data on fuel costs. The bureau would then be in a position to suggest the changes necessary to reduce the fuel costs at any particular plant. Also, there are many places in which the costs of heating and lighting Government buildings can be materially reduced by studying the system employed. It is only through the efforts of experts trained in fuel engineering, especially when from a disinterested branch of the service, that such facts can be brought out and economies effected.

For these reasons it is believed that a special investigation with a view to increased efficiency in the heating and power plants in the public buildings of the United States should be authorized, such work to be in charge of the Bureau of Mines and conducted with the advice and coöperation of the various bureaus and departments interested.

RECENT COURT DECISIONS

Digested by A. L. H. STREET

"Power" Judicially Defined—Under the Kansas Workmen's Compensation Act, defining a "factory" as a place where "power" is used in manufacturing, etc., the term "power" does not include hand power, being confined to energy developed by machinery. (Kansas Supreme Court, Menke vs. Hauber, 160 Pacific Reporter, 1017.)

Compelling Removal of Dangerous Wires—A recent decision of the Vermont Supreme Court interprets the laws of the state relating to the Public Service Commission as giving that board power to entertain and pass upon an application of a telegraph company to compel an electric company to remove its high-power wires from dangerous proximity to the telegraph company's wires. Such an order is not violative of the constitutional provisions guaranteeing the enjoyment of one's property. (Western Union Telegraph Co. vs. Burlington Traction Co., 99 Atlantic Reporter, 4.)

NEW PUBLICATIONS

PUBLICITY FOR PUBLIC SERVICE CORPORATIONS

This booklet is the substance of an address given by Ivy L. Lee, before the annual convention of the American Electric Railway Association at Atlantic City, N. J., Oct. 10, 1916. The work embraces publicity as applied to public-service corporations, fundamental axioms in dealing with the public, the fundamental purpose of a publicity policy, the first necessity in sound public relations, where courtesy must first begin, the personal attitude of a management, the qualities of a manager, a sound policy for a public utility company and the great value of advertising. The author suggests that a man who goes into a policy of publicity must believe absolutely that he is right and that he can justify his policy upon the theory that "truth loves open dealing." The subject is clearly presented in a very readable way and should be of interest not only to those connected with public-service utilities, but also to the public in general. The booklet is issued by the author, 61 Broadway, New York, N. Y.

ENGINEERING OF POWER PLANTS. By Robert H. Fernald, M. E., A. M., Ph. D., and George A. Orrok, M. E. McGraw-Hill Book Co., Inc., New York. Cloth; 586 pages; 6x9 in.; 310 illustrations. Price, $4.

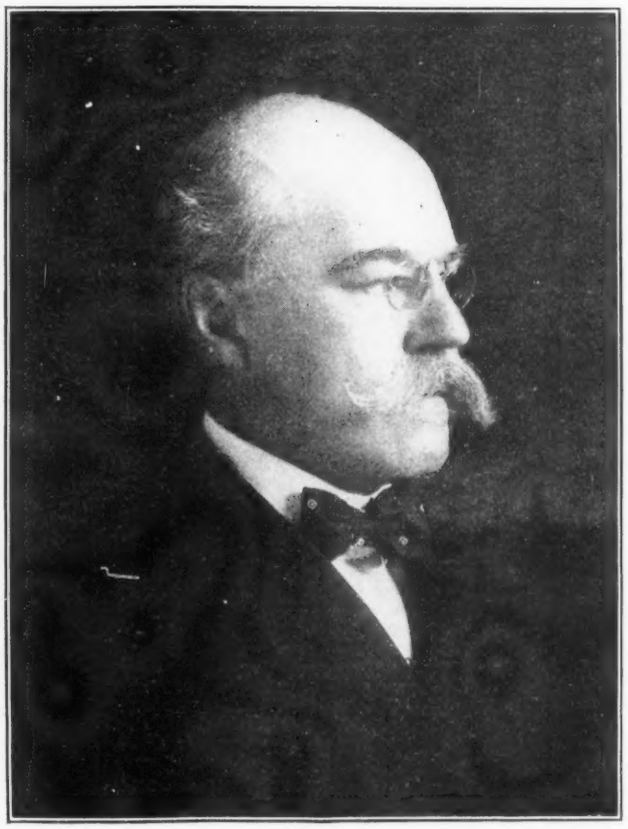
This, like a number of other books that have appeared in recent years, is the bringing together, in a form for convenient classroom use and for the assistance of the engineer, of the notes that have served the author in professional work. In this case, however, Professor Fernald has called into collaboration, in Mr. Orrok, a man who for years, as mechanical engineer of the New York Edison Co. and as a consulting engineer, has dealt with the practical application of the subjects taught, and who brings to the task not only a knowledge of that which should be taught, but a fund of precedents and data from successful practice. Notwithstanding its collegiate and professional purposes, the material is most simply and understandingly presented, with very little mathematics and that of the simplest order.

A definite idea of the plan and scope of the book may best be gained by an enumeration of the subjects of the 26 chapters into which it is divided: Sources of Energy; The Steam Engine; Electric Generators and Motors; Foundations; Condensers; The Steam Boiler; Chimneys and Mechanical Draft; Smoke and Smoke Prevention; Boiler Auxiliaries; Piping; Coal and Ash Handling; The Steam Power Plant; Variable Load Economy; Cost of Power; Hints on Steam Plant Operation; Power Transmission; District Heating; The Power Plant of the Tall Office Building; The Power Plant of the Steam Locomotive; Fuels; Internal Combustion Engines; Producer Gas and Gas Producers; Comparative Efficiencies and Operating Costs for Different Types of Installation; Compressed Air; Refrigerating Machinery, and Hydraulic Power. Fifteen of these chapters have lists of problems to be solved by the student. The book is replete with charts and tables and aims to bring to the student a realization of the fact that engineering, although based on the exact sciences, is not itself an exact science. It requires on the part of the successful engineer a natural fund of common sense and the application of engineering judgment, a realization of the fact that accuracy may mean within 20 per cent. and not the seventh place to the right of the decimal point. Many cost figures are given, but it is advised that they be used with caution on account of local conditions and market variations. Steam turbines are treated in the chapter on The Steam Engine, and notwithstanding their encroachment, the authors say that "the power for rolling mills, blast furnaces, mines and water-works, mine hoisting, air and ammonia compressors, etc., will be furnished by the piston engine for a long time, and engines of similar types will still be used for mill work, where belt or rope driving is preferred. It is only fair to say that in the last five years all the builders of large-sized engines for land work have practically gone out of business, although many engine builders still remain in the field. It is noticeable that only the builders of the higher-class engines and the lower-class engines of the medium and small sizes remain in the field."

The chapter on Cost of Power contains the operating figures of many stations and will be read with interest by power-plant owners and engineers. Outside of its announced purpose, the book is one of the best for the operative engineer who is concerned with the executive and administrative side of his occupation that we have yet seen.

OBITUARY

Alfred Blunt Jenkins died at his home in Llewellyn Park, West Orange, N. J., Friday morning, Dec. 29, 1916, at the age of 68 years. He was born in Boston in 1848, the son of Nathaniel and Mary Tucker Jenkins. In 1872 he and his brother Charles formed a partnership under the title of



ALFRED BLUNT JENKINS

Jenkins Brothers, to continue the manufacture of valves and mechanical rubber goods, a business founded by their father. Later, on acquiring his brother's interest, Alfred B. changed the form of the organization into a company under the same title, distributing a portion of the stock among his employees. The enterprise was built up on the recognized merit of the product and the personality back of it. The broad and liberal attitude of management is well known, but the personal charities and beneficence of the deceased are little known and unheralded, being considered by him as personal matters bearing their own reward in their doing. During the last two months he gave large sums to Dr. Troudeau's Memorial Hospital, Saranac Lake, N. Y., the Manhattan Eye and Ear Hospital, the New York Association for the Blind, and equipped the X-ray operating room of the Orange hospital. In

the founding of the Orange Free Library he took a lively and substantial interest. He was a veteran of the Civil War and a respected "comrade." He is survived by a widow and one daughter, Mrs. Farnham Yardley. The funeral services, held at his late home at 2:30 Sunday and conducted by Rev. Charles T. Walkley, were attended by a large number of his employees and friends.

PERSONALS

Otto Kurtz has succeeded A. Westermeyer as auditor of the Sheboygan Electric Co., of Sheboygan, Wis.

Emil A. Bechstein, for a number of years general manager of the Sandusky (Ohio) Gas and Electric Co., has been elected second vice-president of the company.

Frank B. Jewett has been appointed chief engineer of the Western Electric Co., succeeding C. E. Scribner, who retires to become consulting engineer for the company.

Raymond H. Smith, formerly manager of the Jackson Light and Traction Co., Jackson, Miss., is now vice-president and general manager of the Sheboygan Electric Co., Sheboygan, Wis.

G. A. Ungar, former representative of the SKF Ball Bearing Co. in Cleveland, Detroit and Pittsburgh, has been appointed technical manager and chief engineer of the company.

S. B. Taylor, sales manager of the SKF Ball Bearing Co., of Hartford, Conn., has been appointed vice-president of the company, succeeding F. B. Kirkbride, who remains on the Board.

L. O. Gordon, formerly connected with the sales department of the General Electric Co. and more recently manager of the Valparaiso Lighting Co., is now manager of the Jackson Light and Traction Co., Jackson, Miss.

E. A. Geoghegan has accepted the position of chief mechanical engineer of the Unaversol Superheater Co., of Detroit, a Michigan corporation recently formed to manufacture superheaters for power and industrial purposes.

Gilbert H. Reilly, of the Dearborn Chemical Co., has been made district manager of the company with offices at Syracuse, N. Y. On Dec. 28 a fare-thee-well dinner was tendered to him at the Marlborough Hotel, New York City, in which thirty of his good friends participated.

B. A. Brennan, formerly contract manager of the Westinghouse Machine Co. and more recently sales manager of the power department of the Bethlehem Steel Co., has resigned the vice-presidency of the Mercantile Trust Co., of St. Louis, to accept the presidency of the Citizens Co., Inc., of Baltimore.

P. A. Staples, formerly vice-president and general manager of the New Jersey Power and Light Co., Dover, N. J., and second vice-president of the Sandusky (Ohio) Gas and Electric Co., has been elected vice-president and general manager of the Binghamton (N. Y.) Light, Heat and Power Co., succeeding S. H. Dailey, who has been made general manager of the public utilities operating in Lexington, Ky.

Arthur H. Young, who for some years has been supervisor of Labor and Safety at the Illinois Steel Co., Chicago, took up his new duties as director of the American Museum of Safety, New York, Jan. 1. The Department of Labor and Safety of the Illinois Steel Co. stands today as high in efficiency and the securing of successful results as probably that of any industrial organization in America. Freedom from all labor troubles and the splendid spirit of coöperation existing in the personnel of the company attest to the success of this department as achieved under Mr. Young.

ENGINEERING AFFAIRS

The New England Water-Works Association will hold its annual meeting in Boston, Mass., on Jan. 10, with headquarters at the Hotel Brunswick.

The Steel Treating Research Club will hold a meeting on Wednesday evening, Jan. 10, in the rooms of the Detroit Engineering Society. Dr. Wheeler P. Davey, of the Research Laboratory, General Electric Co., will lecture on the "Detection of Flaws in Metal Castings by Means of the X-Rays," illustrated with lantern slides.

The American Institute of Electrical Engineers will hold its next meeting in Pittsburgh on Jan. 12 with headquarters at the Fort Pitt Hotel. The meeting will be devoted to a discussion of "Braking Electric Vehicles by Regeneration," using the energy generated on down grades to apply brakes to the vehicles. The paper on this subject will be presented by R. E. Hellmund, of the Westinghouse Electric and Manufacturing Co. The session of the Institute will be held in the English Room of the Fort Pitt Hotel in the evening, preceded by an informal dinner.

MISCELLANEOUS NEWS

Steam Pipe Explosion Fatal—Contrary to reports in the daily press, the death of three men and injuries to several others aboard the Old Dominion liner "Princess Anne," while off Lambert's Point, were not due to a boiler explosion, but to the failure of a 5-in. steam line. Failure may have been precipitated by water-hammer.

The Westinghouse Electric and Manufacturing Co. has purchased a tract of land with a deep water frontage on the Delaware River at Essington, Penn., just outside Philadelphia, where it is proposed to erect a large works to handle the greatly increased business of the company. Exclusive of war contracts, unfilled orders are now on the books amounting to over $31,000,000.

Shortage of Natural Gas in Ohio—It is reported that the natural-gas supply has been failing so rapidly in the State of Ohio that the Public Utilities Commission of the state has found it necessary to order the gas companies to discontinue the service to all industries so that the supply may be conserved for as many years as possible for domestic consumption. This is made all the more difficult on account of the coal shortage, which makes it impossible for a change over to be made on short notice.

BUSINESS NOTES

Stephenson & Nichols, Monadnock Building, San Francisco, have been appointed by the Terry Steam Turbine Co., of Hartford, Conn., as its representatives in northern and central California and northern Nevada.

The Lagonda Manufacturing Co., of Springfield, Ohio, announces the opening of a new branch office, in the McCormick Building, Chicago, Ill. J. E. Chubb, formerly with the Griscom-Russell Co., is in charge as district sales manager. The company's business in this territory was formerly in charge of the Chicago Engineer Supply Co.

The Blackburn-Smith Corporation, 105 West 40th St., New York, through purchase of the filter department of James Beggs & Co., has acquired all patents, records, patterns and rights for the manufacture and sale of the Blackburn-Smith feed-water filter and grease extractor and the Blackburn-Smith sewage-ejector system. The engineers and agents previously identified with these products have been taken over in a body by the new corporation and will in the future work under the direction of Irvin Kaufman, vice-president, who has supervised the manufacture, sale and installation of the Blackburn-Smith filter for the past ten years.

The H. A. Rogers Co., 87 Walker St., New York, states that, due to the European war, the brand of gage glasses handled by it will, until further notice, be sold at "prices extant on date of shipment." In a circular letter the company states that import orders require six months to execute, but that orders to be filled from the New York stock will be shipped as and when it is possible to do so. Gage glasses are hard to obtain and may be harder to get, therefore it is suggested that all users conserve their own interests by not being wasteful.

A Consolidation—The new "Shoot Your Ashes" system, known as the American steam jet conveyor is manufactured and installed by the American Steam Conveyor Corporation, 326 West Madison St., Chicago. The new firm combines the systems, patents and personnel of the Girtanner-Daviess Engineering and Construction Co., of St. Louis, Mo., and the Griffin Engineering and Construction Co., of Elkhart, Ind. Both of these firms had been manufacturing successfully for several years; both had several hundred ash-removal systems in successful operation. The two systems, while similar in principle—that of removing ashes through a pipe by the suction of a steam jet—differed somewhat in operation. Each had its limitations. With the consolidation of the two systems into one, the American steam jet conveyor, practically every possible condition of installation can be met.

THE COAL MARKET

Boston—Current quotations per gross ton delivered alongside Boston points are as follows:

ANTHRACITE

	Circular		Individual	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago
Buckwheat	$3.60	$3.10	$6.00@6 50	$3.00@3.50
Rice	3.10	2.50	5.50@5.75	2.60@3.00
Boiler	2.95	....	5.00@5.25	.....
Barley	2.85	2.25	4.20@4.60	2.45@2.60

BITUMINOUS

Prices per gross ton for Boston delivery:

	F.o b. Mines, All-Rail Rate $2.60 to Boston		Alongside Boston Water Coal	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago
Clearfields	$4.75@5.60	$2 50@3.25	$8.25@8.75	.....
Cambrias and Somersets	5.00@6.00	2.85@3.25	8.50@9.00	.....
	F.o.b. Hampton Roads		On Cars, Boston	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago
Pocahontas and New River	$7.00@7 50	$2.85	$9.25@9.75	$6.00@6.25

New York—Current quotations per gross ton f.o.b. Tidewater at the lower ports are as follows:

ANTHRACITE

	Circular		Individual	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago
Buckwheat	$2.75	$2.75	$5 50@6.00	$2.47@2 75
Rice	2.20	2.25	5.00@5.50	1.95@2.25
Barley	1.95	1 75	3.75@4.00	1.50@1.75

Quotations at the upper ports are about 5c. higher.

BITUMINOUS

	Sbuth Amboy		Port Reading	Mine Price	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	Jan. 6, 1917	One Year Ago
Clearfield	$7.50@8.00	$3.90@4.15	$7.50@8.00	$6.00@6 25	$2.25@2.60
South Forks	7.50@8.00	4.05@4.30	7.50@8.00	6.00@6.25	2.50@2.75
Nanty Glo	7.50@8.00	4.05@4 30	7.50@8.00	6.00@6.25	2.50@2.75
Somerset	7.50@8 00	3.75@4.15	7.50@8.00	6.00@6.25	2.00@2.75
Quemahoning	7.50@8.00	3.75@4.15	7.50@8.00	6.00@6.25	2.00@2.75

Philadelphia—Prices per gross ton f.o.b. cars at mines for line shipment and f.o b. Port Richmond for tide shipment are as follows:

	Line		Tide	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago
Buckwheat	$2.00	$1.50	$2 90	$2.25
Rice	1.25	.85	2.15	1.75
Boiler	1.10	....	2.00	....
Barley	1.00	.50	1.90	1.25

Birmingham—Current prices per net ton f.o.b. mines:

	Mine-Run	Washed Mine-Run	Washed Nut
Pratt	$4.00@4.25	$4.50	$4.50
Big Seam	3.50@4.00	$3.75@4.25	$3.75@4 25
Black Creek	4.00@4.25	4.50	4.50
Carbon Hill	.....	4.00	4.00
Cahaba	4.00@4.25	4.50	4.50

Pittsburgh—Price of steam coal per net ton f.o.b. mine, Pittsburgh district:

	Jan. 6, 1917	One Year Ago
Slack	$4.95@5.05	$1.50@1.75
Mine-run	5.00@5.25	1.50@1.75
¾-in.	5.00@5.25	1.60@1.85

Add 40c. per ton for freight charge to Pittsburgh.

Chicago—Current prices per net ton f.o.b. mines:

	Springfield	Clinton and Sullivan Cos.
Steam lump	$4 00@4.50	$4.00@4.25
Egg	4.00@4.50	3 75@4.25
Nut	4.00@4.50	3.75@4.25
Screenings	4.00@4.25	3.50@4.25
Mine-run	4.00@4.25	3.50@4.50

	Williamson and Franklin Counties	Saline and Harrisburg	West Virginia Smokeless
Lump	$4.00@4.50	$4 00@4.50	$5 50@6.00
Egg	4.00@4.50	4.00@4.50	5.50@6.00
Nut	.....	4.00@4.25	.....
No. 1 nut	4 00@4.50	.....	.....
No 2 nut	4.00@4 50	.....	.....
No. 3 nut	4.00@4.25	.....	.....
No. 1 washed	4.00@4.50	.....	.....
No. 2 washed	4.00@4 50	.....	.....
Mine-run	4.00@4.25	4.00@4.25	5.25@5.75
Screenings	4.00@4 25	4.00@4 25	.....

Hocking lump, $1; splint lump, $4.25

St. Louis—Price per net ton f.o.b. mine:

	Williamson and Franklin Counties		Mt. Olive and Staunton		Standard	
	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago	Jan. 6, 1917	One Year Ago
6-in. lump	$3 50	$1.50	$3.50	$1.25	$3.50	$1.10@1.15
2-in. lump	...	1.35	...	1.15	3.25@3 50	.90@1.00
Steam egg	...	...	3.50	...	3.25@3.50	.....
Mine-run	3.50	1.10	3.25	1.00	3.00@3.25	.85@ .90
No. 1 nut	...	1.20	...	...	...	.90@1.00
2-in. screenings	3 25	.80	3 25	.75	3.00@3.25	.70@ .75
No. 5 washed	3.25	.80	3.00	.70	3.00	.70

Williamson-Franklin rate St. Louis, 72½c.; other rates, 57½c.

PROPOSED CONSTRUCTION

Calif., Escondido—The San Diego Consolidated Gas and Electric Co., San Diego, plans to construct a power transmission line from Oceanside to Escondido, about 25 miles.

Calif., Vallejo—Bureau of Supplies and Accounts, Navy Department, Washington, D. C. is receiving bids for 1,500 ft. rawhide belt lacing, 108 brass bibb cocks, composition pipe fittings, slocknuts, brass pipe and 1000 lb. copper tubing, under Schedules Nos. 580 and 583, for Navy Yord at Mare Island, Calif.

Fla., Panama Park—City plans to install an electric-light system.

Fla., Pensacola—Bureau of Supplies and Accounts, Navy Department, Washington, D. C. is receiving bids Jan. 16, for two 12-in. "K" transformers, under Schedule No. 562 for Navy Yard at Pensacola.

Iowa, Centerville—The Iowa Southern Utilities Co. plans to rebuild its power plant.

Ky., Booneville—City voted in favor of $10,000 bonds to install an electric-light plant.

Md., Baltimore—The Consolidated Gas, Electric Light and Power Co. has received authority from the Public Service Commission to issue $900,000 bonds for extending and improving its plant. A. S. Loigeaux is electrical engineer.

Minn., Akely—City plans to construct an electric-light plant. D. Walker, secretary of the Red River Lumber Co., is interested.

Minn., Brookpark—City plans to construct an electric-light system. R. P. Allen, Pine City, manager of the Eastern Minnesota Power Co., is interested.

Minn., Glenwood—Glenwood Electric Light, Heat and Power Co. granted permission to build an electric-light plant. R. M. Smedstad, village recorder.

Minn., Minneapolis—The Minneapolis General Electric Co. plans to construct a 200x375-ft. plant at Chestnut Ave. and 16th St. About $500,000.

Minn., Wabasha—The Wisconsin-Minnesota Light and Power Co., Eau Claire, plans to construct a substation in Wabasha.

Mont., Harlem—W. L. Saul has purchased the electric-light plant from R. J. Moore, and plans to extend and improve same.

N. Y., New York—(Borough of Brooklyn)—Bureau of Supplies and Accounts, Navy Department, Washington, D. C., is receiving bids for 2000 lb. of round cotton wick packing, 50 electric heaters, lighting and power wire, 3800 ft. triplex cable, 7000 ft. single conductor wire, 110,000 ft. telephone wire, 120,000 ft. ½-in. galvanized steel conduit, 7400 lb. lead pipe, under Schedules Nos. 570, 575, 577 and 577 and 578, for Navl Yard at Brooklyn.

N. Y., Taughannock Falls—The Ovid Electric Co. plans to rebuild its transmission line next summer.

N. D., Gwinner—Local business men have organized a company to construct an electric-light plant in Gwinner.

N. D., Napoleon—The Napolean Light and Power Co., recently incorporated with a capital stock of $15,000, plans to construct an electric-light and power plant. W. W. Noddings, E. A. Wellan and others of Napoleon are interested.

Ohio., Cleveland—The National Acme Co., 212 East 131st St., plans to construct a power plant. About $16,000.

Okla., Laverne—Laverne Electric Light, Ice and Power Co., recently incorporated with a capital stock of $30,000, plans to construct a plant. W. H. Lewis and others are interested.

Okla., Norman—City plans to install an electric-light plant.

Ont., Galt—Hydo-Electric Commission plans to enlarge its main substation on Dickson St. E. B. Fewings is chairman.

Penn., Philadelphia—Bureau of Supplies and Accounts, Navy Department, Washington, D. C. is receiving bids for 3400 ft. of unlined linen fire hose, 150 steel boiler tubes, galvanized steel or iron pipe straps, under Schedules Nos. 569, 574 and 577, for Navy Yard at Philadelphia.

R. I., Newport—Bureau of Supplies and Accounts, Navy Department, Washington, D. C., is receiving bids Jan. 16 for 1000 lb. of copper tubing, under Schedule 563 for Navy Yard at Newport.

Tex., Dallas—Council plans to install an electric-light system in West Dallas.

Tex., Garwood—J. J. Cooper and associates plan to construct and electric-light and power plant.

Tex., Mahank—Council plans to install an electric-light system.

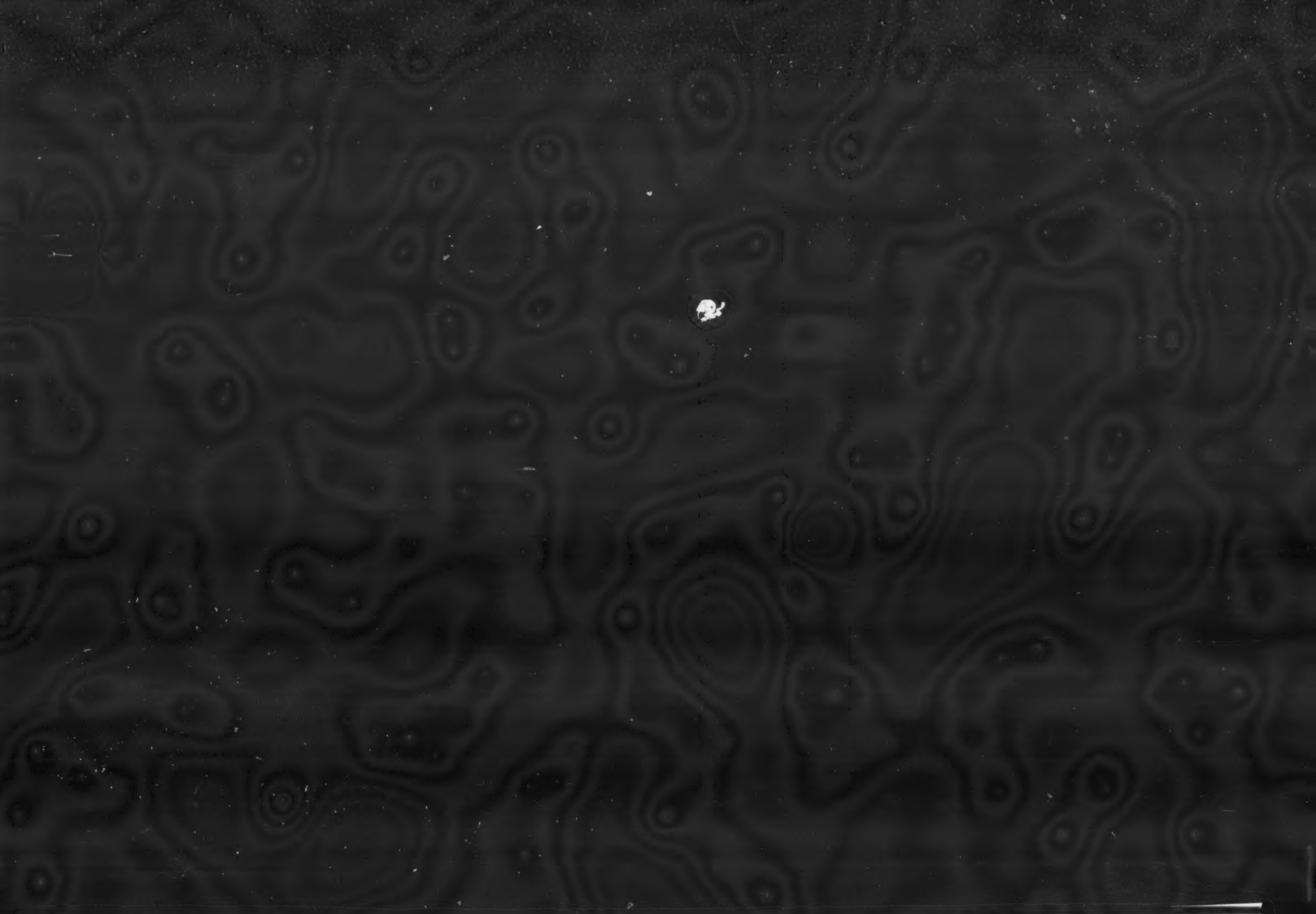
Tex., Mart—City Commission granted a franchise to the Mutual Electric Co. to construct an electric light plant. L. Harlan is interested.

Va., Norfolk—(Official)—Bureau of Yards and Docks, Navy Department Washington, D. C., is receiving bids Jan. 22 for an electric light and power system for the structural shop at the Navy Yard, Norfolk.

Va., Suffolk—Planters' Nut and Chocolate Co. plans to construct a one-story boiler house. C. F. Nevins, architect.

Wash., Seattle—City plans to improve its electric light and power plant.

Wis., Amery—Apple River Milling Co. has plans prepared for a power plant. J. C. Jacobson, 743 Security Bldg., Minneapolis, architect.

Wis., Eden—The Wisconsin Gas and Electric Co., Kenosha purchased the Eden Electric Light and Power Co., also received franchise to extend its transmission lines to Marblehead.

Wis., Random Lake—City plans to vote on bond issue to construct a lighting system.